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Final Technical Report
J6020

EVALUATION OF WEAR RESISTANCE OF FOURTEEN GEAR MATERIALS

June 1970

Engineering Mechanics Division
IIT Research Institute
10 West 35th Street
Chicago, Illinois 60616

Final Technical Report, Project J6020
(Period: June 1965 to March 1970)

Prepared by

W. J. Courtney

Under Contract No. NAS5-9590

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For

National Aeronautics and Space Administration
Goddard Space Flight Center
Greenbelt, Maryland

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FOREWORD

This program was conducted for the NASA Goddard Space Flight Center under Contract No. NAS5-9590, IITRI Project J6020. This report summarizes the work performed on the subject contract during the period of 22 June 1965 to 21 March 1970. Mr. Charles E. Vest acted as NASA's Technical Officer. It was conducted in the IIT Research Institute's Engineering Mechanics Division with project engineering direction by Messrs. W. J. Courtney, J. J. Farrell and S. Guzder.

Respectfully submitted,

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VACUUM TESTS OF INSTRUMENT SIZE GEARS -- EVALUATION OF SELECTED GEAR MATERIALS

By W. J. Courtney

SUMMARY

Eight gear materials are evaluated for wear characteristics. Brief descriptions are given of the four-square gear test fixtures, master gear arrangements, and instrumentation for the measurement of wear rates. Involute profile (368) and cycloidal profile (32) gears were used.

Four test series were conducted. The first series consists of involute gears tested at a constant speed (1800 rpm) and constant torque load (20 oz-in). These tests include 14 atmospheric and 42 vacuum tests. Evaluations with 224 involute gears indicate that the nitrided nitralloy and 440C stainless steel material combinations operate for the prescribed time (720 hr) in both laboratory and vacuum atmosphere. Martin hard coated 7075T6 aluminum alloy is paired with nitrided nitralloy and in another combination with C1085 steel (silver plated and lubricated with in situ molybdenite (MoS_2). These combinations operate for the prescribed time in vacuum only. In laboratory ambient evaluations, nitrided nitralloy and Phosphor bronze combined with 15 percent MoS_2 matrix material operates for the prescribed time.

The second series of tests is performed on 32 carburized C1020, nitrided nitralloy, and 440C stainless steel gears of cycloidal profile. Six tests are conducted in vacuum and two in atmosphere. The results indicate that these gears will not operate for 720 hr in vacuum at either 10 or 20 oz-in. torque load and 1800 rpm without accumulating greater than 10 percent wear. The atmospheric tests show similar results at 20 oz-in. torque load and 1800 rpm.

The results of the third series are inconclusive because of the nitrogen enrichment of the nitrided nitralloy gears which caused chipping. Forty nitrided nitralloy and 40 stainless steel 440C gears of involute profile were tested at 10, 20 and 30 oz-in. torque load and speeds of both 900 and 1800 rpm.

In the fourth series 16 unlubricated gears (nitrided nitralloy and 440C stainless steel) and 44 lubricated gears (nitrided nitralloy and Martin hard coated 7075T6) are compared using 3 boundary film type lubricants. Lubricated gears are subjected to 1800 rpm, 20 oz-in. torque load; unlubricated gears are operated at 900 or 1800 rpm, and at 10, 20 or 30 oz-in. torque load.

INTRODUCTION

The wear characteristics of selected gear materials are evaluated in terms of applicability for vacuum use. Material selection is based on the requirement of unlubricated or dry film (MoS_2) lubricated gears. The literature shows that very hard materials or materials with hard surfaces have extended wear lives. Also, hard surfaced materials are primarily intermetallic compounds, characteristically resistant to cold welding in the space environment. To meet the need for a corrosion resistant material, a through-hardened 440C stainless steel is evaluated. A light anodized 7075T6 aluminum alloy is used as a basis for comparison. A through-hardened beryllium copper alloy is included for its nonmagnetic, electrical, and thermal conductance properties. The eight materials selected are listed in Table I.

These materials are evaluated for wear rates and tooth surface degeneration using gears in four-square gear testers at speeds of 900 and 1800 rpm under 10, 20, and 30 oz-in. torque loads, which correspond to contact stresses of ~30 000, 60 000 and 90 000 psi. Each test is continued until the gear tooth profile is reduced by ~10 percent or until 720 hr of continuous

Table I

GEAR MATERIALS SELECTED FOR
WEAR EVALUATIONS

Material	Description
I	Carburized C1020 Steel (case depth 0.002 to 0.003 in.)
II	Nitrided Nitralloy 135 Modified Steel (case depth 0.002 to 0.003 in.)*
III	Beryllium Copper Alloy 25 (heat treated to Rc41-44)
IV	Deep Anodized 7075T6 Aluminum (Martin hard coated, case depth 0.002 to 0.003 in.)
V	440C Stainless Steel (heat treated to Rc55-60)
VI	Phospor Bronze (15 percent MoS_2 matrix material)
VII	C1085 Steel (heat treated to Rc50, silver plated 0.0001 in. with E_3C Molykote film)
VIII	Light Anodized 7075T6 Aluminum Alloy

*The case depth was between 0.005 and 0.007 in. for the second series of nitrided gears.

running elapses. The time limit is based on economic considerations. This testing program constitutes a first order selection and evaluation of materials. The following comparisons are noted:

- At 20 oz-in. torque load and 1800 rpm, only the nitrided nitralloy vs 440C stainless steel gears (involute profile) ran for 720 hr without accumulating greater than 10 percent wear in both atmospheric and vacuum environmental tests.
- Beryllium copper alloy 25 is undesirable for use as an involute gear material at 20 oz-in. torque load and 1800 rpm, either in the laboratory or vacuum environment.
- Both light anodized 7075T6 aluminum and Phospher bronze with 15 percent MoS_2 matrix possess inadequate mechanical properties as an involute profile to support the required load.
- The cycloidal profile performs unsatisfactorily with all three materials, 440C, C1020 and nitralloy.

DESIGN AND CONSTRUCTION OF THE TEST FACILITY

The first major phase of the program was the design and construction of a test facility for evaluating materials using instrument size gears under both vacuum and laboratory ambient environmental conditions. Since there were 224 gears and a total of 14 different material combinations to be evaluated, it was felt that a test facility capable of simultaneously testing 32 gears of different material combinations would be adequate for completing the program in a maximum period of 7 mo, based on the requirement that each test be conducted for a maximum of 720 hr. This period does not include the changeover and setup time between tests. Two separate testing apparatus were designed, one for use in the laboratory atmosphere environment (Figure 1) and the other for the vacuum environment (Figure 2).

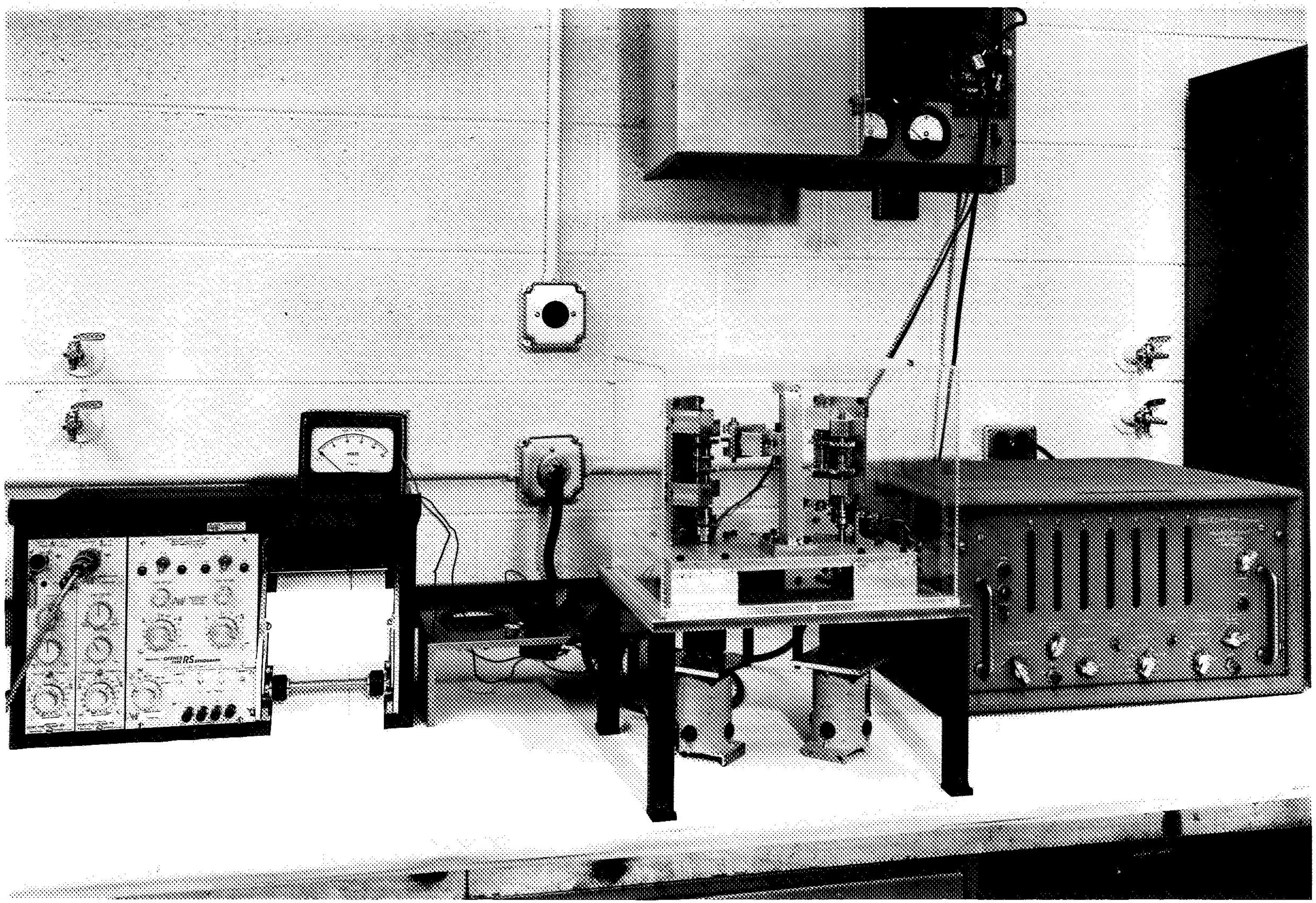

Figure 1 Apparatus for Laboratory Ambient Gear Wear Tests

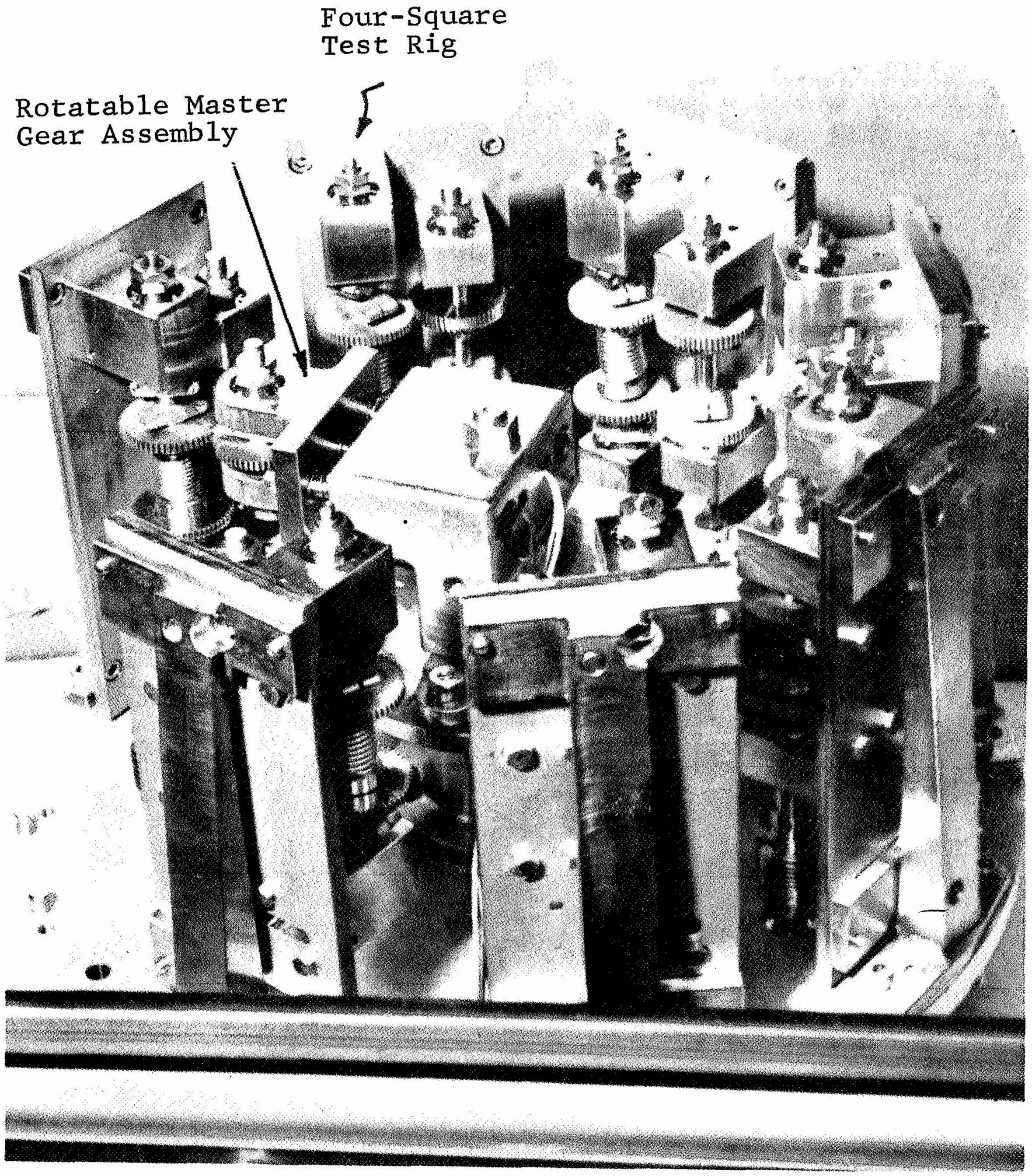

Figure 2 Test Apparatus for Vacuum Environment

Vacuum System

Of all the space environmental parameters affecting the operation of mechanical components, the "hard vacuum" is one of the most influential. Reproduction of the exact density, molecular flux and composition of the environment is essential for meaningful tests but if the fundamental interreactions can be predicted, these factors can be accounted for. However, to reduce the interaction of mechanical wear surfaces with the impinging gas species to a negligible level, it is necessary to reduce the molecular flux to a level at which the monolayer absorption is very long compared to the time for one revolution of a gear. Thus, for a gear speed of 1800 rpm, a chamber pressure of 5×10^{-8} torr is used to provide space-vacuum simulation. The complete vacuum system used (Figure 3) consists of:

- A 14-in. diam and 12-in. high stainless steel vacuum chamber with four view ports, one 14 in. Wheeler flange, three electrical feedthroughs, one mechanical rotary feedthrough, and a 6 in. pump manifold with two roughing valves.
- Two Varian Vac Sorb roughing pumps.
- A Varian water cooled titanium sublimation pump.
- One 500 liter/sec Vac Ion pump.

The vacuum chamber has its bottom plate fabricated with eight cup-shaped depressions to accommodate the magnets of each four-square test fixture comprising one-half of the magnetic drive. The other half is mounted on separate motors assembled outside and directly under each test rig. A manual mechanical rotary feedthrough in the center of the bottom plate is used to position the master gear mechanism for periodic wear measurements.

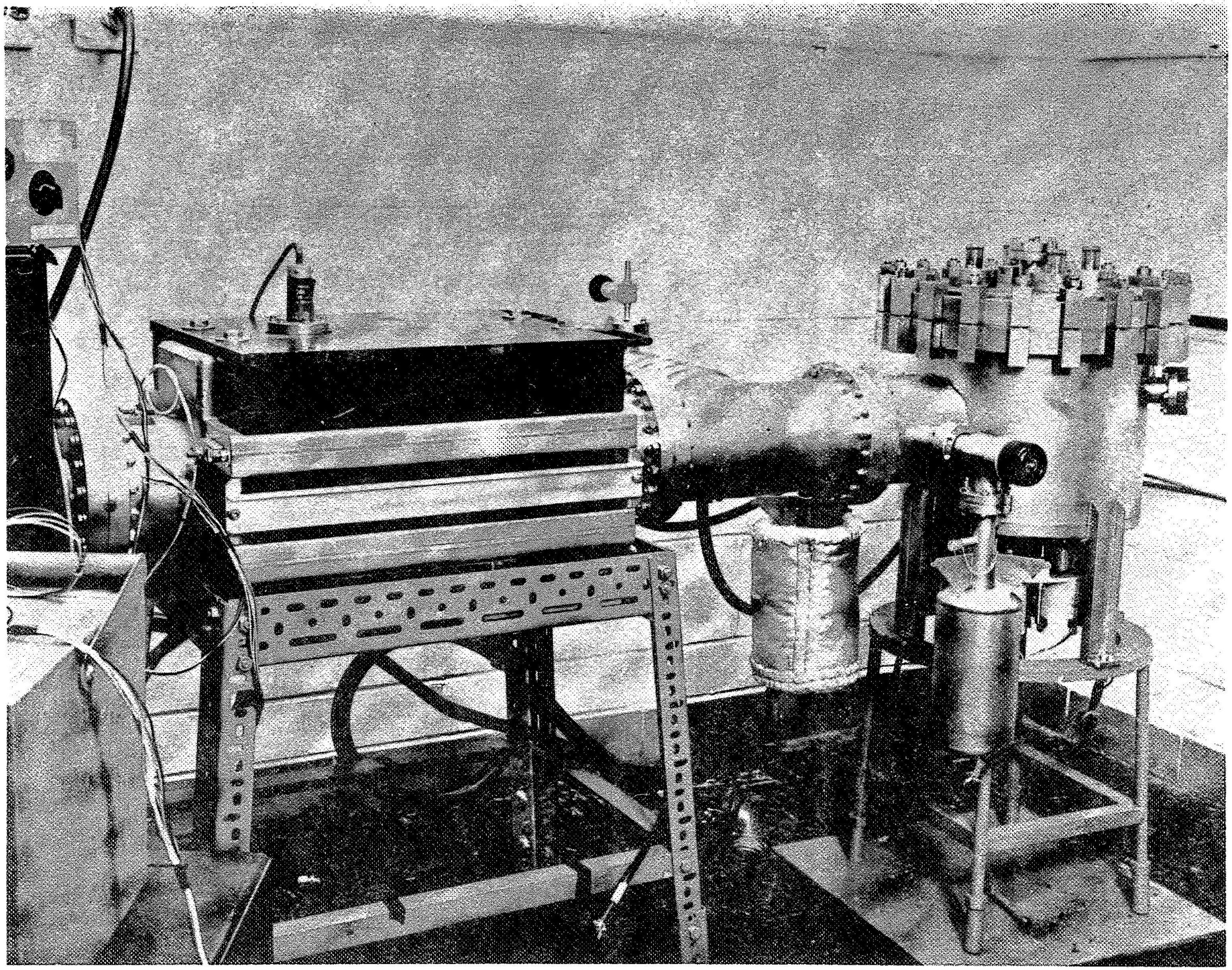

Figure 3 Ultrahigh-Vacuum System for Gear Wear Studies

Four-Square Test Rig

A four-square gear testing configuration was selected to fulfill the contract requirement that all materials be tested under constant load and speed. The utilization of the four-square arrangement is considered ideal for testing gears in a vacuum environment since it permits operation of the gears under the desired loads without imposing strenous requirements on mechanical vacuum chamber penetrations.

A four-square gear tester is a test apparatus which allows a torque load to be applied to two sets of test gears. The four-square test rig designed and used on this program is shown in Figure 4. The desired torque load is obtained with the amount of twist applied to the torque spring used to couple the two short shafts together. These shafts are supported in the larger bearing blocks which contain a double set of size R-4 precision ball bearings. A set of 55 teeth test gears is mounted on these two shafts. These gears are mated with a set of 56 teeth test gears mounted on the main drive shaft supported in the small bearing blocks which contain single bearings. The bearings used in the four-square rig are New Hampshire SR4 PB11 ball bearings with retainers of Salox M material. The bearings are axially loaded as discussed in Appendix A.

The original design called for the main drive shaft to be connected by a flexible torsional coupling to the shaft containing the driven magnet, as indicated in Figure 4. Rotation of the flexible torsional coupling could be used to indicate the torque required to drive the rig. However, preliminary tests indicated that the arrangement caused fretting corrosion of the shafts. Therefore, it was replaced by a stiff spring allowing for slight misalignment in the assembly. This reduces but does not eliminate the vibration induced fretting noted in two locations: between the inner races of the bearings and the shafts, and a lower degree

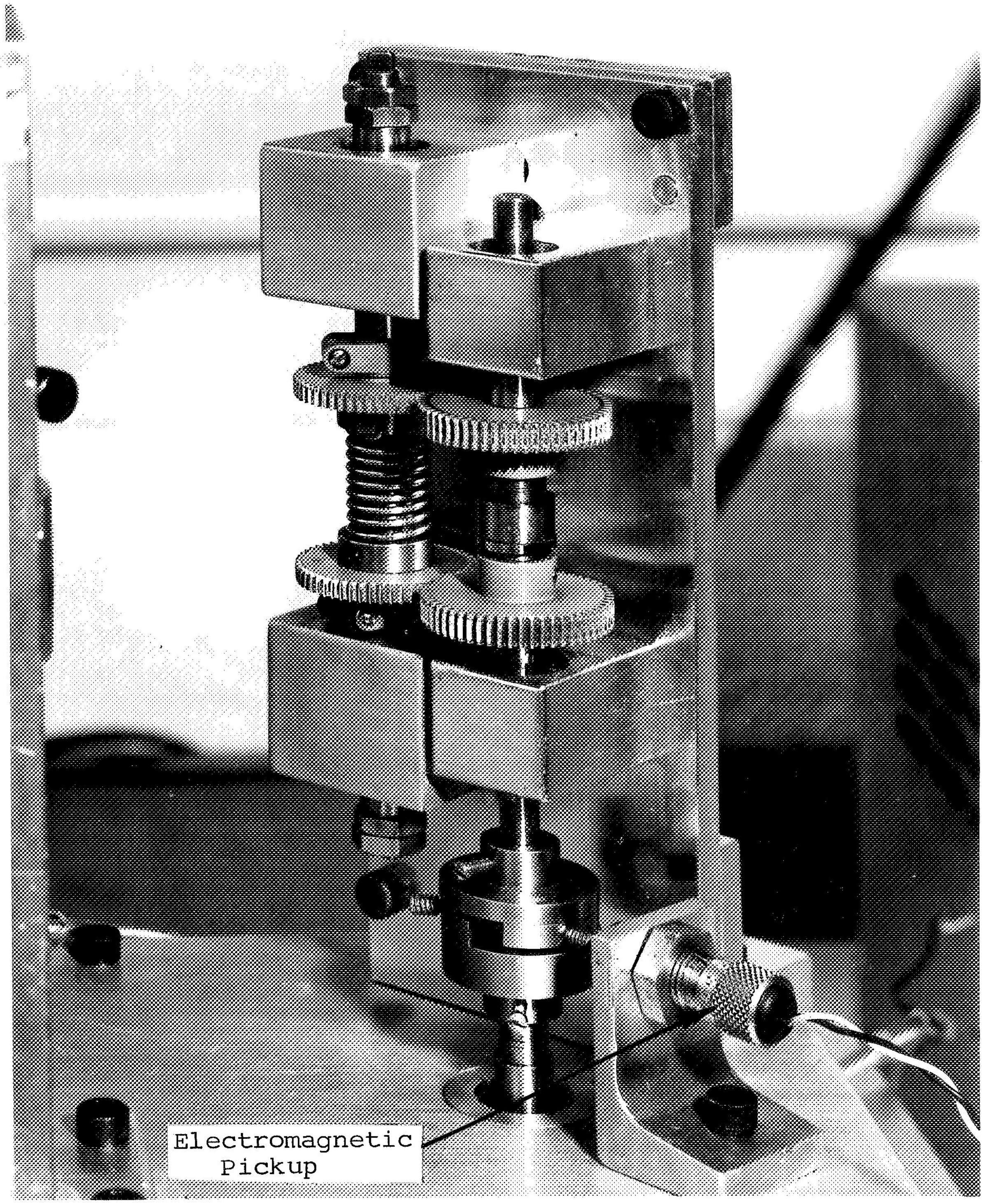

Figure 4 Four-Square Test Fixture

between the gears and the shafts (under the collar clamp). This fretting is due to the vibrational loading induced by the bellows coupling.

The life of the bearings is, in many cases, limited to one test. When a gear set is removed from the fixture the bearing is destroyed due to the extreme axial loads required to remove the shaft from the bearings. Preliminary tests also indicate the bearing blocks must be pinned in place to maintain a fixed center distance. By actual vacuum test it is found that the center distance required must be equal to the sum of the pitch radii of the mating test gears, plus the total composite error (0.0005 in.) plus 0.002 in. This distance is 0.001 in. greater than recommended by AGMA for quality No. 12 gears and is necessary because of the thermal expansion of the test gears which occurs in vacuum. In addition to the four-square test fixtures and drive motors, both apparatus include a master gear mechanism. The four-square test rigs in the vacuum apparatus are positioned around the master gear mechanism such that all test gears are at the same radial distance from a vertical reference centerline which coincides precisely with the centerline of the rotation of master gear mechanism.

Master Gear Arrangement for Gear Wear Measurement in Vacuum

A spring-loaded master gear* arrangement is used in the test facility. This arrangement (Figure 5) enables periodic approximation of the degree of wear on the test gears in vacuum. Most gear manufacturers utilize this technique in determining the precision of gears. The arrangement basically consists of a spring-loaded ultraprecision master gear or a precision gear of known profile mating with a test gear. On revolving the combination and precisely

*These so-called "master gears", as discussed later, are gears with rectangular teeth.

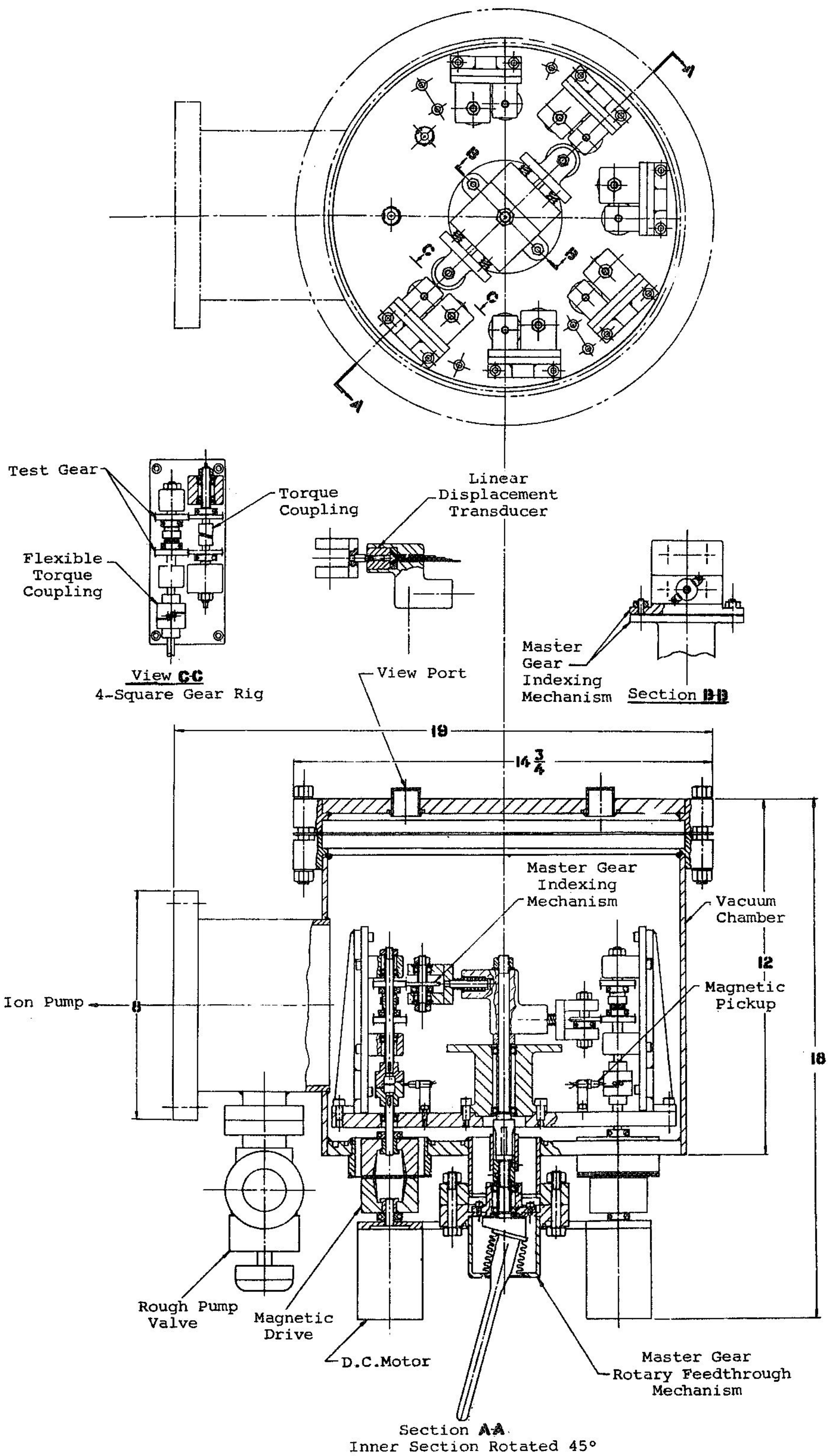

Figure 5 Gear Wear Test Apparatus in Vacuum

measuring the variations in the center distance between the master gear and the test gear, accurate measurements of tooth-to-tooth composite tolerance (TCE) and total composite tolerance can be made.**

The master gear is mounted in a U-shaped bracket precisely sliding and spring-loaded within another rotary bracket which can be moved vertically and also revolved to bring the master gear into contact with any of the test gears (Figure 6). While mating with a slowly revolving test gear, the variations in the center distance are measured by a linear variable displacement transducer (LVDT) which senses the movements of the U-shaped bracket supporting the master gear. The purpose of this arrangement is, therefore, twofold: (1) to make periodic TCE measurements, and (2) to determine the amount of wear on the tooth profile by measuring changes in a reference distance between the master gear and the test gear. In Figure 5 the individual LVDT's are mounted on a rotary feedthrough mechanism. A precision indexing arrangement is also provided on the master gear mechanism such that the master gear can be accurately indexed for repeated monitoring of each test gear profile. Although this method of monitoring gear wear in the vacuum chamber appears quite straightforward, many problems and solutions to each are discussed in the following writeup.

In the initial design, ultraprecision master gears of the same diametral pitch as the test gear were selected. However, this was found satisfactory for measuring wear of up to only 1 or 2 percent of the test gear, because of interference between the

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**Tooth-to-tooth composite tolerance is defined as the allowable variation in center distance when a gear is rotated (in tight mesh with a master gear) through any increment of 360°/N (N = number of teeth in gear under inspection). Total composite tolerance is defined as the allowable variation in center distance when a gear is rotated (in tight mesh with a master gear) one complete revolution. (This includes the effects of variations in active profile, lead, pitch, tooth thickness and run-out.)

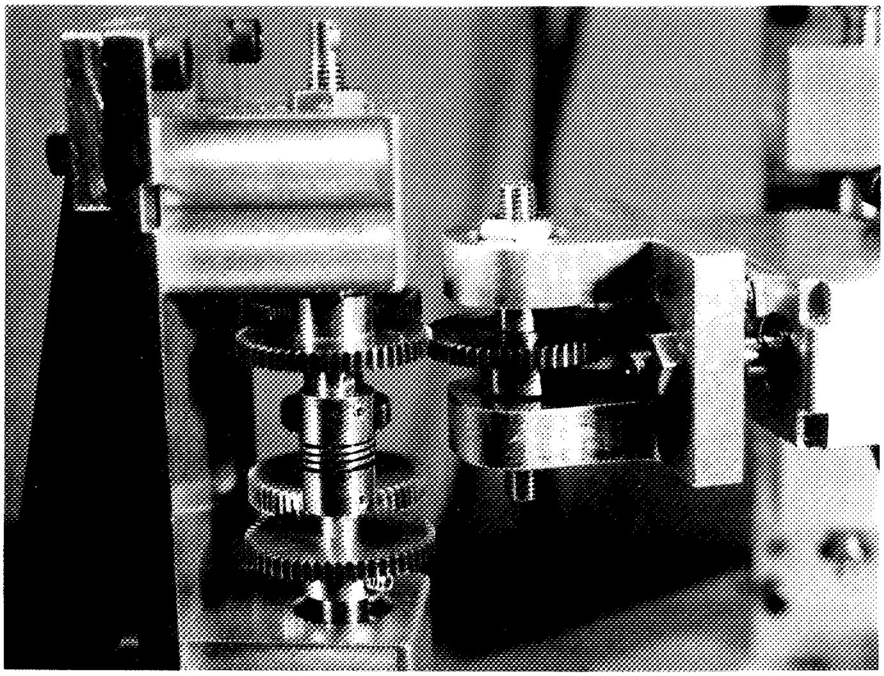


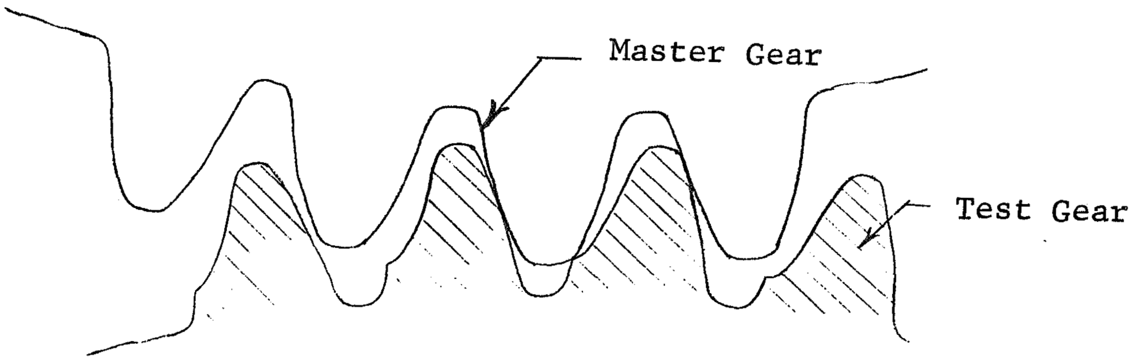
Figure 6 Master Gear Arrangement for Laboratory Ambient Environment

addendum portion of the master gear teeth and dedendum portion of the test gear when a large percentage of wear occurred (Figure 7a). Hence, the master gear teeth had to be modified. This was achieved by grinding the addendum of the master gear teeth down to the pitch line and reducing the sides of the teeth to a rectangular shape (Figure 7b) to remove further interference. Figure 8 illustrates the standard over-pin measurement used to maintain wear in the atmospheric tests.

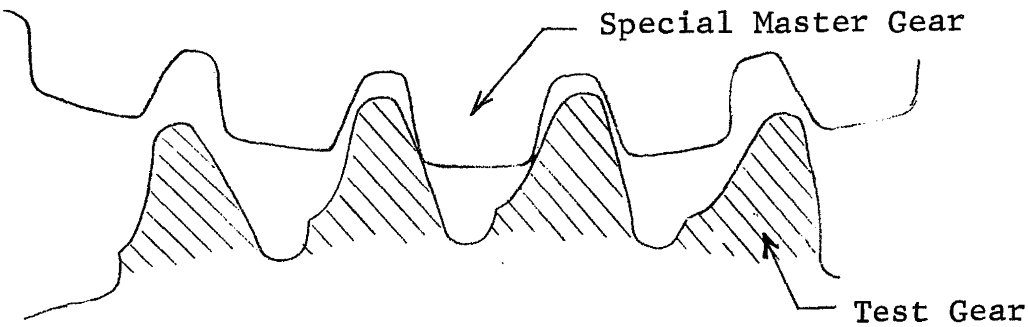
The modification of the master gear teeth resulted in a rectangular tooth thickness equal to the tooth width (0.0327 in.) at the pitch radius, whereby during measurements the rectangular tooth makes contact with the test gear at or below its pitch radius. In other words, the changes in center distances were reflected only by the amount of wear on the dedendum portion of the teeth on the test gear. Careful studies of actual wear profiles on several gears revealed that this approach was not accurate enough for interpreting percentage wear at the pitch radius by measurements of changes in center distances because different gears showed appreciably different wear rates below the pitch radius, for the same percentage wear at the pitch radius. This led to further modification of the master gear requiring a larger tooth width (0.0375 in.) so that the actual wear rate could be interpreted with better accuracy for measured changes in the center distance.

One problem, encountered only with the vacuum gear test apparatus, which seriously affected the wear rate measurements was the change in physical dimensions of various sections of the test apparatus due to thermal expansion during testing. This problem, and the necessary steps taken to overcome it, can be best explained by referring to Figures 9a, b and c.

The top view of the master gear mechanism is shown in Figure 9a where C is the fixed center of one of the test gears and A is the fixed center of the test apparatus around which the master gear is spring-loaded toward the center of the test gear. The dimension



(a) Interference Between Addendum of Master Gear Teeth and Dedendum of Test Gear



(b) Mating of Modified Rectangular Master Gear Tooth with Test Gear

Figure 7 Gear Wear Measurement Technique in Vacuum

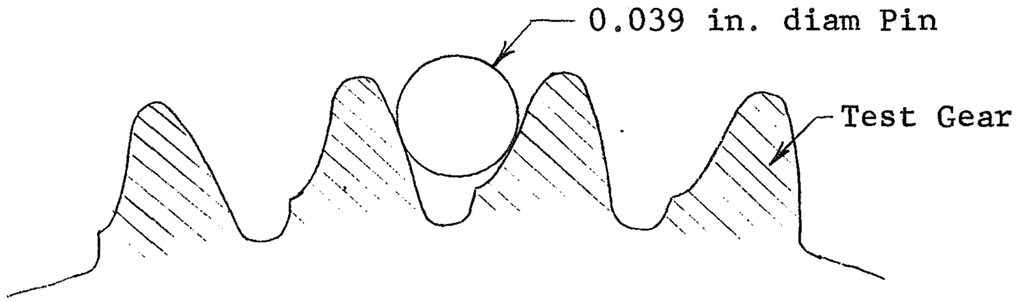


Figure 8 Gear Wear Measurement Technique in Atmosphere

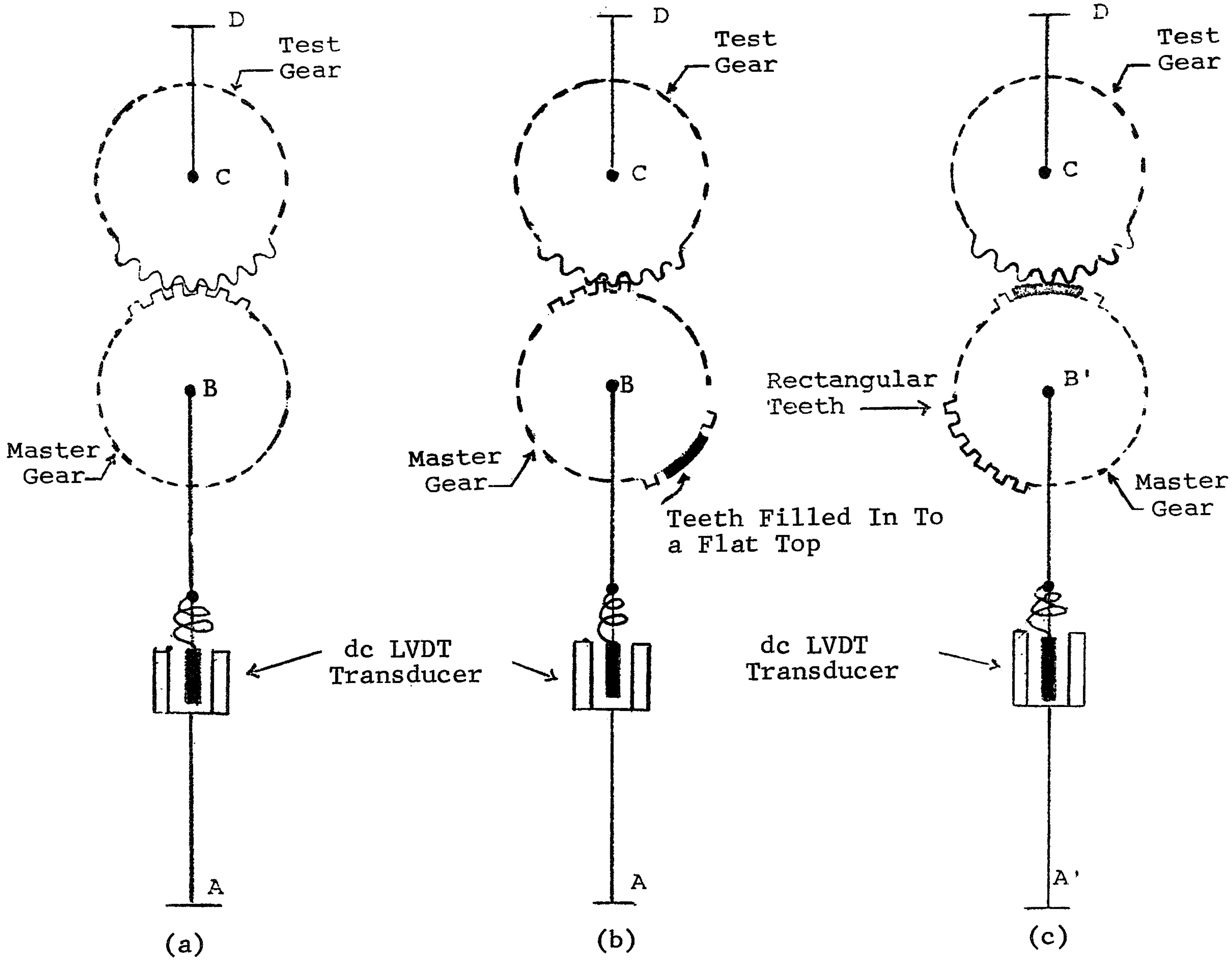


Figure 9 Wear Measurement Using Master Gear

AB is determined by the amount of wear on the test gear teeth. The change in dimension AB is, therefore, the measure of the amount of wear on the test gear and is indicated by the change in output of the direct current LVDT transducer. In other words, change in center distance BC is measured instead of change in AB. This approach would work provided AC were always constant. However, due to the lack of convective heat transfer in the vacuum chamber, a considerable amount of heat generated is transferred to different sections for the test apparatus, establishing thermal gradients that cause unpredictable thermal expansion. Thermal expansion of the base plate of the apparatus on which all four-square test fixtures are mounted would reflect a change in dimension AC and cause serious error in the wear measurement. This problem was detected by serious drifts in transducer output showing that AB was changing even when the test gear did not have any wear. In order to compensate completely for any change in physical dimensions due to temperature rise, a different method of measurement was used. This method is shown in Figures 9b and c. Here the master gear has, in addition to its rectangular teeth, a portion of its circumference without any teeth. Figure 9b shows the normal measurement as in Figure 9a where the output of the transducer indicates AB. However, in Figure 9c the toothless portion of the master gear mates with the outside diameter of the test gear indicated by the transducer as A'B'. The important thing to note here is that although AB and A'B' are subject to drifts due to thermal expansion causing AC to drift, the difference AB-A'B' at any time is independent of changes in AC, and is only dependent on the amount of gear wear. This is based on the assumption that no wear occurs on the outside circumference of the test gear. This approach was adopted on all tests conducted in vacuum environment, so that instead of making just one measurement of AB, one additional measurement, A'B', was made for every test gear. The difference AB-A'B' is used as an indication of the amount of wear.

Another problem arose when the equilibrium temperature of several components in the vacuum test rigs was measured using a contact thermocouple. These measurements indicated body gear temperatures of ~180°F and upper bearing block temperatures of 145°F. However, these temperatures are only approximate because of the influence of contact pressure on the actual reading. Calculation of the gear temperature increase based on the measured change in gear dimensions indicated that the gears were closer to 275°F. The location of bearing blocks with respect to the gears is shown in Figure 40 of Appendix A. A complete description of the gear wear measurement technique is included in Appendix B.

Gear Wear Measurement in the Laboratory Environment

Periodic wear measurements are made of test gears in the laboratory test rig as shown in Figure 8. A 0.039 in. diam pin arrangement is employed for quick measurement of wear by a simple over-pin measurement with a micrometer. A calibration curve is generated by plotting over-pin measurements for a number of worn gears against the actual percentage wear occurring at the pitch diameter, as determined with the toolmaker's microscope technique. By this method, the final determination of the percentage wear which occurs during the gear wear tests is made with a toolmaker's microscope after the gears are removed from the test rigs. This is accomplished with a microscope with a micrometer table movement having an accuracy of 0.0001 in., in two perpendicular directions. The accuracy or repeatability of these measurements is better than ± 0.0002 in., which is equivalent to ± 0.6 percent wear at the pitch radius. It was found that a decrease in the over-pin measurement of 0.012 in. is equivalent to ~10 percent wear at the pitch line of the gear.

The selection of a 10 percent (0.0032 in.) decrease in the width of the test gear at the pitch radius as an indication of 10 percent reduction in tooth profile was made after the preliminary results from the initial wear tests were available for correlation checks. These results show that 10 percent reduction in tooth profile cannot always be accurately predicted. The degree of wear is determined by actually measuring the tool profile after the test is completed.

GEAR MATERIAL SPECIFICATIONS, PROCUREMENT AND FABRICATION

Of the eight materials selected for evaluation and presented in Table I, beryllium copper alloy 25 (material III) was originally specified as Sintered Aluminum Powder (SAP). However, due to the unavailability of a sintered aluminum powder gear material with the desired oxide content, the substitution was made.

Involute gears were used in three sets of evaluation testing. The original contract specified 14 different combinations using the eight materials. Subsequent contract modifications were concerned with further testing of involute gears; 80 gears fabricated from nitrided nitralloy and 440C stainless steel were used in one subset; and 60 gears fabricated from nitrided nitralloy, 440C stainless steel and Martin hard coated 7075T6 aluminum alloy comprised another subset. The 7075T6 gears and their mating nitrided nitralloy gears were lubricated.

During the test of the 32 cycloidal profile gears, 8 carburized C1020, 12 nitrided nitralloy and 12 stainless steel 440C gears were evaluated. These included the following combinations:

nitrided nitralloy vs 440C stainless steel
440C stainless steel vs carburized C1020
nitrided nitralloy vs carburized C1020
nitrided nitralloy vs nitrided nitralloy

The order of testing the involute and cycloidal gears was as follows: the 224 involute gear set was evaluated first; the 32 cycloidal gears were tested second; 80 involute nitralloy with 440 stainless steel followed; and last, the 60 involute gears (nitralloy, 440C and 7075T6 aluminum) gears were tested.

Specifications

The specifications followed for manufacture of involute and cycloidal gears are given in Table II. The cycloidal gears do not possess tolerances equivalent to those of the involute gears evaluated during the study. The primary reason for this difference is the lack of adequate precision grinding equipment for final grinding of the cycloidal profile following heat treatment. Therefore, the cycloidal gears are not ground after heat treatment, but instead are tested with the as-hobbed and heat-treated tolerances.

The problem of interference due to insufficient backlash required the inactive face of each 0.125 in. wide cycloidal gear to be lapped to provide backlash. The original unlapped and lapped profiles of a C1020 cycloidal gear are shown in Figure 10.

Material Procurement and Gear Fabrication

During the initial procurement of materials, a review of existing manufacturing procedures (1965) for precision gears, AGMS No. 12 quality, indicated that gears of all originally specified materials (except those of Materials VI and III (SAP) would require finish grinding after heat and/or surface treatments. At the time we located three companies with adequate facilities:

Aero Gear and Tool Corp.
Little Ferry, N. J.

Riley Gear Corp.
North Tonawanda, N. Y.

Equitable Engineering Co.
Detroit, Mich.

Material problems encountered in the gear fabrication are of particular interest especially when the various hardened surfaces are considered. The problem areas will be considered individually.

Table II

SPECIFICATIONS FOLLOWED FOR MANUFACTURE OF INVOLUTE AND CYCLOIDAL GEARS

Item	Involute Gears Specification: AGMA Classification No. 12	Cycloidal Gears Specification: British Cycloidal Standard BS 978, Part 2
Diametral Pitch	48	48
Pressure Angle, deg	20 deg	
Tooth-To-Tooth Composite Tolerance, in.	0.0003	
Total Composite Tolerance, in.	0.0005	
Backlash	Designation "D" (backlash per mesh of 0.0003 to 0.001 in.)	
Number of Teeth	55, 56	55, 56
Face Width, in.	0.187 ± 0.005 0.125 ± 0.005	$0.187 \pm .005$, $0.125 \pm .005$
Pitch Diameter, in.	$1.1458^{+0.0000}_{-0.0007}$, $1.1667^{+0.0000}_{-0.0007}$	$1.1458^{+.0000}_{-0.0007}$, $1.1666^{+.0000}_{-.0007}$

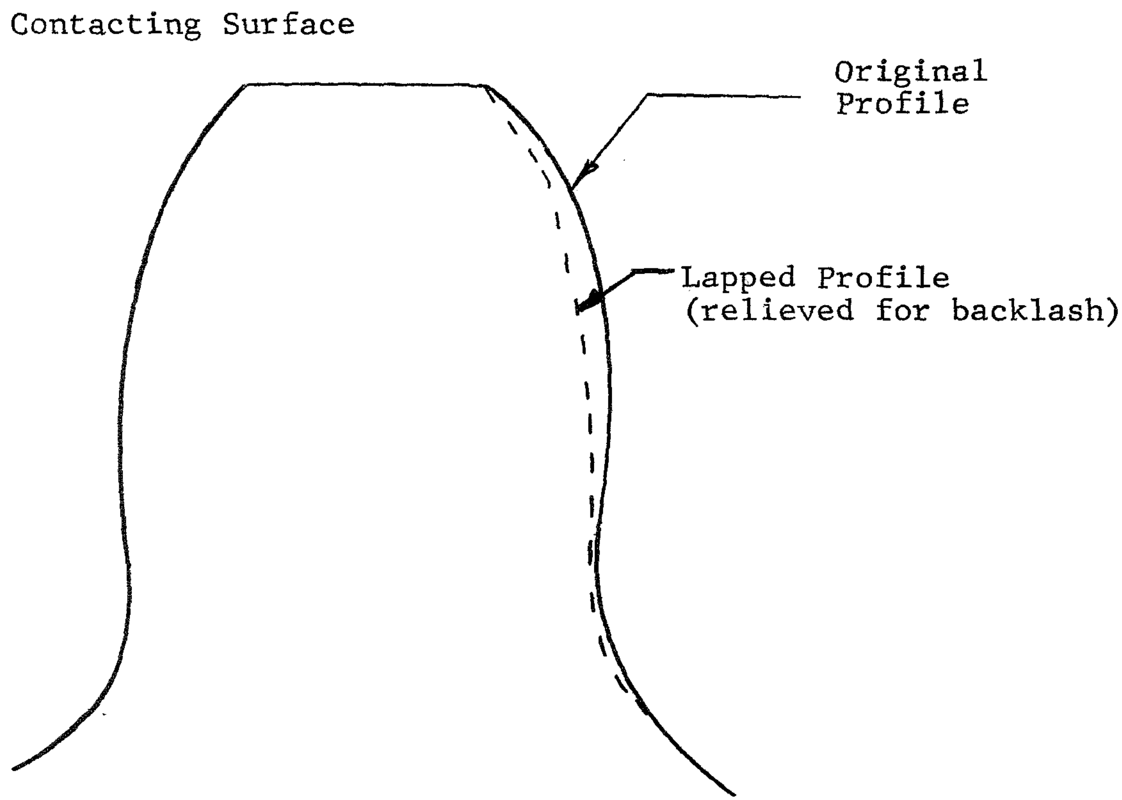


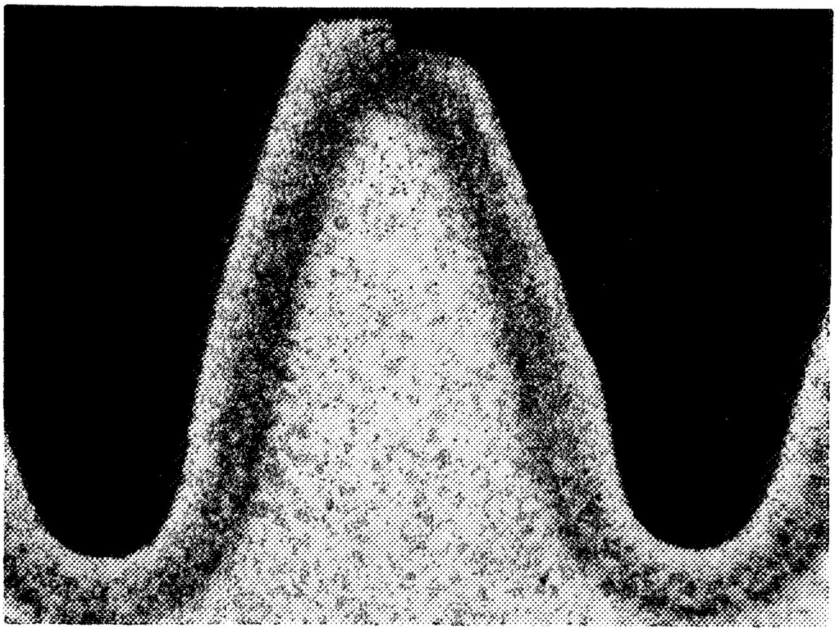
Figure 10 Cycloidal Profile of 0.125 in. C1020 Gear

Nitrided nitralloy gears. - A problem encountered during the fabrication of the nitrided nitralloy gears for the second series of involute profile tests occurred when photomicrographs of these gears indicated a marked difference between second series gears and the nitrided gears used for the first series of involute gear tests (Figures 11 and 12). The light colored area between the outer thin white layer and the characteristic dark nitrided matrix which appears in Figures 12a and b was not presented in the first series (Figure 13).

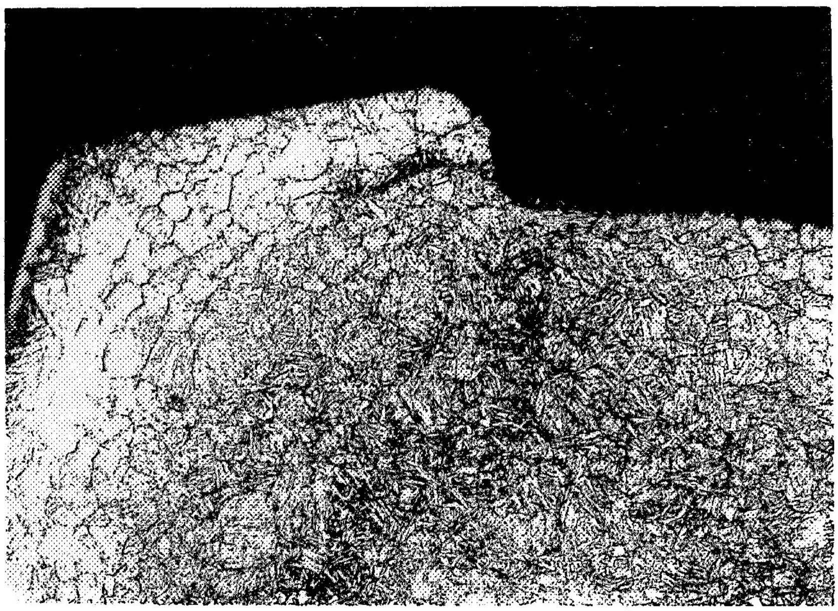
Discussion with IITRI's metallurgists and commercial heat treaters (Lingbergh Heat Treat) revealed that, in production nitriding, variations in the percentage of dissociated ammonia present during the initial nitriding cycle can cause this condition. The percentage of dissociated ammonia is manually controlled, and fluctuations of from 5 to 8 percent are not uncommon. Such a grey layer is normally caused by a low percentage of dissociated ammonia during the first cycle which allows a nitrogen enriched layer to form below the normal white coat. As mentioned, this type of formation is primarily a function of the level of dissociated ammonia present during the first cycle of the two-stage nitriding process; but is also influenced by the duration of each cycle. However, all these parameters are influenced by the amount of specimen surface exposed in the furnance.

The company that performed the heat treating (L & R Metal Treating of New Jersey) uses a two-stage flow process. The first cycle consists of a 5 hr period with 15 to 25 percent dissociated ammonia present (manually adjusted at 1/2 hr intervals) at 975°F. The second stage consists of 3 hr at 1050°F and ~83 percent dissociated ammonia.

It should be remembered that nitriding is not an exact and well-defined technique using existing production equipment and practice. Good consistency can only be achieved by careful control of the critical parameters. This was confirmed by the fact that

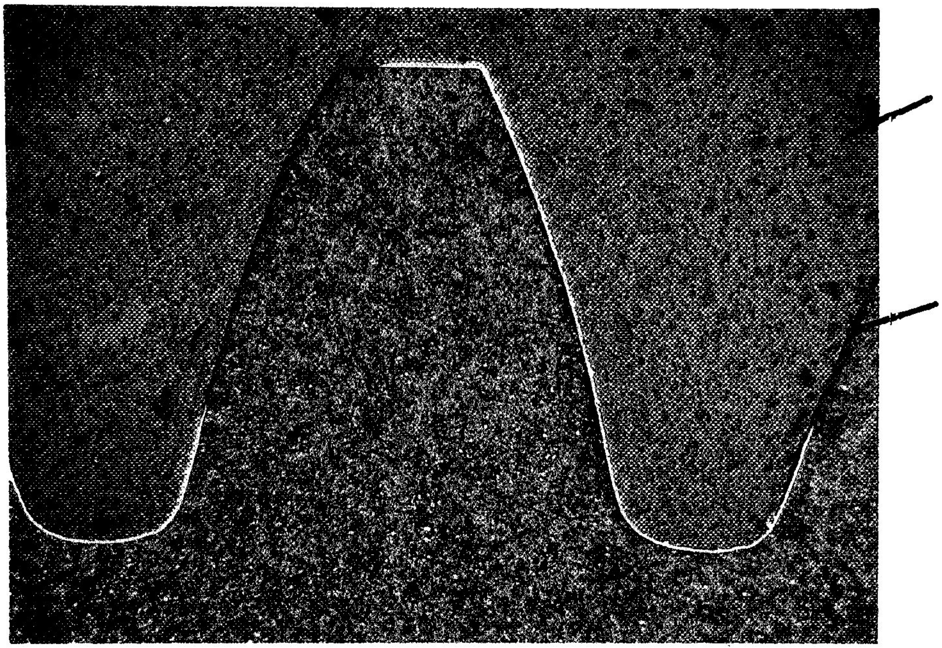


Etch: 2 percent Nital, 50X Magnification
(a) Nitrided Gear Tooth with Chipped Tip



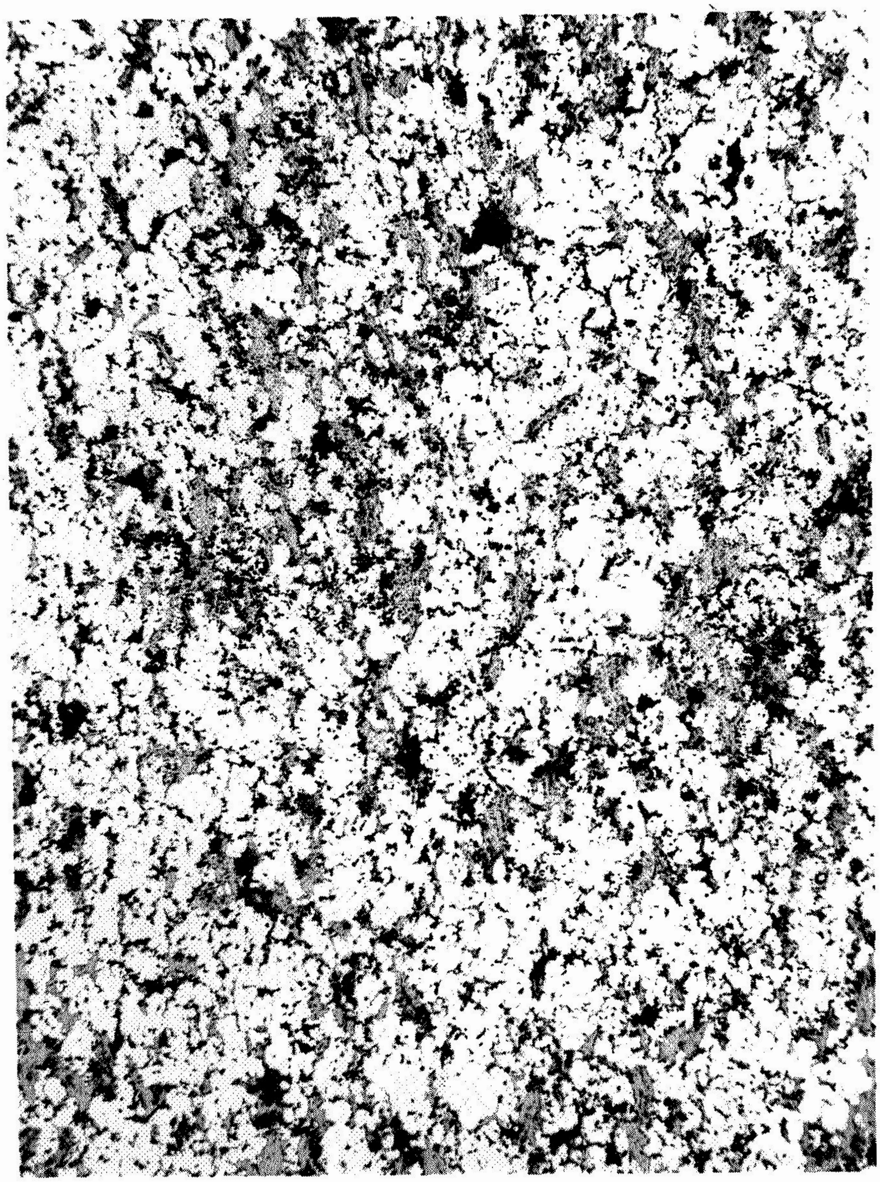
Etch: 2 percent Nital, 250X Magnification
(b) Tip of Chipped Nitrided Gear

Figure 11 Second Test Series Showing Chipping of Nitrided Nitralloy Involute Gears



Etch: 2 percent Nital, 50X Magnification

Figure 12 First Test Series Showing Approximately Correct Thickness of Nitriding



(100X Magnification)

Figure 13 Microstructure of Phosphor Bronze (15 percent MoS_2 matrix) the two series of nitrided gears used in the program were fabricated by the same companies with the same specifications, but different nitrided cases resulted. When the process was controlled more carefully, in the fabrication of a third set of nitrided nitralloy gears, an acceptable case depth resulted.

Martin hard coated 7075T6 aluminum alloy. - These aluminum alloy gears also exhibited erratic results in case depth due to process control problems. In both cases it was found advisable to determine the case depth with sample coupons, both prior to the actual gear surface treatment and as a test coupon during the actual processing.

Sintered materials. - Evaluation of SAP samples indicated they were very close to the desired theoretical density, but lacked sufficient aluminum oxide content. Therefore, as discussed previously, this material was replaced by beryllium copper alloy 25, heat treated to Rc41 - 44.

Figure 13 shows a microstructure of one of the samples. The specimen was diamond polished and unetched. The porosity of the material can be easily seen in the structure. The grey areas indicate pores or places where MoS_2 was pulled out during the polishing process. No attempt was made to estimate the MoS_2 content from the microstructure study; however a fairly good distribution of MoS_2 is seen. Chemical analysis of these specimens showed the MoS_2 content was $\sim$6.90 percent by weight. The theoretical MoS_2 content, assuming 15 percent in the specimen, would be 11.2 percent. Therefore, this means the specimens were impregnated to only 9.25 percent MoS_2. Therefore, a Phosphor bronze matrix with a 15 percent (by volume) MoS_2 impregnation was requested.

METHOD OF TESTING

The 224 involute gears are arranged to facilitate the running of each test gear against another gear of a different material. Thus, 14 different material combinations are evaluated. One run is made in atmospheric environment with each combination, and three runs are made in vacuum. Fifty-six such individual tests are conducted using the four-square gear testers previously described.

The studies conducted in vacuum are carried out in an ion pumped vacuum environment at a pressure of less than 5×10^{-8} torr, a speed of 1800 rpm, and a 20 oz-in. torque load; the laboratory tests are conducted at the same speed and torque load.

Prior to initiation of the vacuum tests a series of transducer measurements is made on each test using the procedure described in Appendix B. These initial measurements are used in conjunction with periodic readings taken during the test period to obtain a history of the gear wear process until greater than 10 percent wear occurs at the pitch line of the test gear. The bearings in the four-square gear testers are run for short periods of time in air to provide a small amount of lubrication to the balls and races of the bearing, prior to installation of the new size R-4 bearings.

During the test period the armature current of the dc drive motor is monitored on a multipoint recorder, every 15 min. Since the armature current of the permanent magnet dc motors is directly proportional to the torque output of the motors, this monitoring provides a continuous record of torque input to the test rigs. Table III shows the 14 combinations of materials evaluated during this phase of the program in both atmospheric and vacuum environments. These tests are terminated after a total running time of 720 hr accumulates on the test gears or when the wear process reduces the gear profile by 10 percent.

Table III

COMBINATIONS OF MATERIALS EVALUATED

Material	Description
I	Carburized C1020
II	Nitrided Nitralloy
III	Beryllium Copper
IV	Martin Hard Coated Aluminum
V	440C Stainless Steel
VI	Phosphor Bronze (15 percent MoS_2)
VII	C1085 Silver Plated and MoS_2
VIII	Light Anodized Aluminum

Material vs Material	
I	II
II	III
II	IV
II	V
II	VI
II	VII
III	IV
III	VI
III	VIII
IV	VI
IV	VII
IV	VIII
V	VI
V	VII

In the first 68 tests all gears are run at 1800 rpm, a few of the sets are tested with loads of 10 oz-in. but the bulk of the gears are loaded to a torque of 20 oz-in. Tests 69 through 73 are made with cycloidal gears, discussed later in this section.

Tests 78 through 97 are made with involute gears fabricated from nitrided nitralloy and 400 stainless steel. These tests are run at two speeds and three torque loads: 900 and 1800 rpm; and 10, 20 and 30 oz-in. Tests 106 through 109 are made with involute gears fabricated from nitrided nitralloy and 440C stainless steel. Two of these tests are run at 900 rpm with torque loads of 10 and 30 oz-in.; the other two are run at 1800 rpm and 20 oz-in. load. Tests 111 through 121 are made with nitrided nitralloy vs Martin hard coated 7075T6 aluminum alloy. All of these gears are lubricated and run at test speeds of 1800 rpm and 20 oz-in. torque loads. The lubricants are solid films based on MoS_2.

During the tests using cycloidal profile gears the test procedures are the same as those described for the involute gears. However, additional effort is expended to generate a relationship between the transducer readings recorded during the test and actual wear. This effort is required because of the difference between the cycloidal and involute gear profiles and also because the inactive face of the 0.125 in. thick cycloidal gears has to be lapped to provide sufficient backlash. Table IV shows the material combinations, torque loads, speeds and environments used for the cycloidal gear tests.

Experimental Results

The results of the vacuum testing of the 14 gear material combinations (for involute gears) are presented in Table V. The combinations are ranked in descending order of their ability to resist wear. A three-test sample is insufficiently large to derive satisfactory statistical evaluations. However, in the

Table IV

CYCLOIDAL GEAR TEST CONDITIONS

Environment	Material Combination	Load oz-in.	Speed rpm
Vacuum	II vs I	20	1800
	II vs V	10	1800
	I vs V	10	1800
	II vs V	20	1800
	I vs V	20	1800
	II vs II	20	1800
Laboratory Ambient	II vs V	20	1800
	I vs II	20	1800

Table V

SUMMARY OF GEAR WEAR IN VACUUM AT 20 OZ-IN. TORQUE LOAD AND 1800 RPM (INVOLUTE PROFILE)

Material Combination	Test Duration hr	Test Number	Maximum Wear Percent	Material With Maximum Wear
II vs IV	720	15	2.2	Same
	720	17	2.5	Same
	720	19	2.5	Same
IV vs VII	720	27	2.0	Same
	720	28	4.0	IV
	720	26	2.0	Same
II vs V	720	13	3.0	Same
	720	18	3.5	Same
	720	20	3.1	Same
II vs VII	720	37	5.8	Same
	720	38	7.0	II
	393	36	10.0	Same
V vs VII	720	30	10.0	Same
	460	35	9.0	VII
	262	29	12.0	VII
I vs II	720	16	3.5	II
	133	39	3.5	I
VIII vs III	314	58	8.6	VIII
	301	59	11.4	VIII
	337	60	7.4	VIII
IV vs VIII	167	23	8.0	VIII
	167	24	9.0	VIII
	594	25	12.0	Same
IV vs III	133	48	12.2	IV
	137	49	11.3	IV
	337	50	6.1	IV
IV vs VI	29	61	---	Teeth sheared
	26	62	---	off VI
	17.5	63	---	
V vs VI	29	64	---	Teeth sheared
	29	67	---	off VI
	6	68	---	
II vs VI	7.2	46	---	Teeth sheared
	4	47	---	off VI
	119	45	10.0	
II vs III	1.5	40	---	Terminated due
	24	41	---	to increased
	24	42	---	torque
VI vs III	3.5	51	---	Teeth sheared
	1.5	52	---	off VI
	1	57	---	Terminated, torque too high

cases where some of the tests lasted 720 hr and others of the same material combination did not, the wear rate even for the sets that lasted is higher as we read down the table.

Table VI shows the test results for involute gears exposed to laboratory ambient air. As indicated in the table, only three combinations of materials complete the test: nitrided nitralloy vs Martin hard coated 7075T6 aluminum; Martin hard coated 7075T6 aluminum vs C1085 heat treated to Rc50 silver plated with 0.0001 in. of E_3C Molykote film; and nitrided nitralloy vs 440C stainless Rc55-60. These exhibit excellent wear properties in vacuum. The other material combinations fail to run 720 hr before accumulating 10 percent wear. The Phosphor bronze gears, as noted in Table VI, fail prematurely by tooth breakage. Most of the beryllium copper gears show evidence of plastic flow, material transfer, and high wear rates.

Representative gears tested are shown in Figures 14 through 25. Each of the eight materials is shown at least once. Figures 26 and 27 show the torque required to rotate the gears at 1800 rpm as a function of time for the life of the gears. Figures 28 through 30 show the approximate percentage wear as a function of running time for the three material combinations found acceptable in the vacuum tests. As noted, percentage wear is indicated as approximate, since it is a plot of wear as measured with the transducers which are strongly influenced by the wear profile which varies from gear to gear. Wear is rapid during the early portion of the test, increases quite slowly after the run-in period the first 100 hr, then remains relatively constant.

Results of the initial screening tests are also presented in Tables V and VI. Examination indicates that only two of the material combinations tested ran for 720 hr with less than 10 percent wear during the screening tests, namely: nitrided nitralloy vs 440C stainless steel; and nitrided nitralloy vs Phosphor bronze with a 15 percent MoS_2 matrix. Only the nitrided nitralloy vs 440C stainless steel combination completed the tests in both vacuum and laboratory environments.

These results indicate that beryllium copper and light anodized 7075T6 aluminum gears run very poorly in both laboratory ambient and vacuum conditions. The Phosphor bronze 15 percent MoS_2 matrix gears run much better in laboratory atmosphere than in vacuum; however, the Martin hard coated gears do not perform as well in the laboratory environment.

The material combination of nitrided nitralloy vs 440C stainless steel was chosen for further unlubricated testing with expanded speed and torque loads (900 rpm, 10 oz-in.; 900 rpm, 20 oz-in.; 900 rpm, 30 oz-in.; 1800 rpm, 10 oz-in. and 1800 rpm, 30 oz-in.). The results are summarized in Table VII. Since all of the tests resulted in chipped gear teeth, it was determined that the nitrided case depth was much too large.

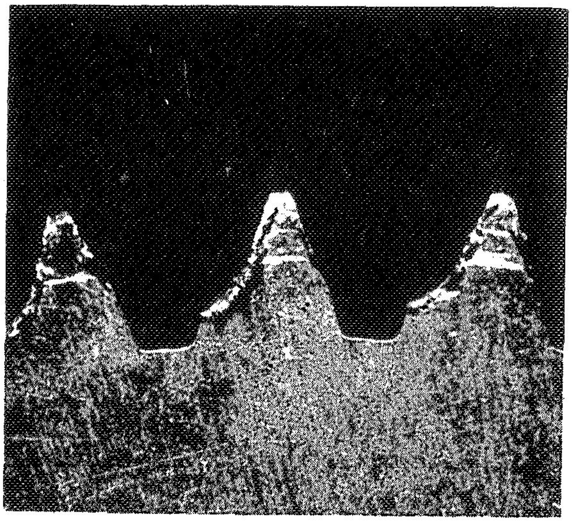
A third set of gears of the same material combination was procured and tested in a vacuum environment. The results are shown in Table VIII. Note that all of the tests show wear in excess of 4 percent in at least one gear.

Further testing was conducted in vacuum with involute gears fabricated from nitrided nitralloy vs Martin hard coated 7075T6 aluminum alloy. The lubricants are coded: A, ultrapure MoS_2, burnished onto the surface of the gears; B, Electrofilm; and C, Dow Corning Company's experimental ultrathin (both B and C are also MoS_2 based lubricants). None of these lubricated gears completed the 720 hr test period and one (Test 116, Table IX) had very low wear when halted at 340 hr. Test 116 was prematurely halted when inordinately irregular wear was observed.

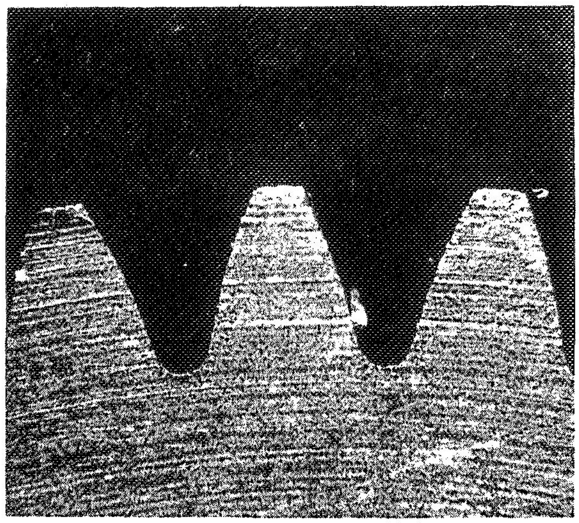
Table VI

SUMMARY OF GEAR WEAR IN ATMOSPHERE AT 20 OZ-IN. TORQUE LOAD AND 1800 RPM (INVOLUTE PROFILE)

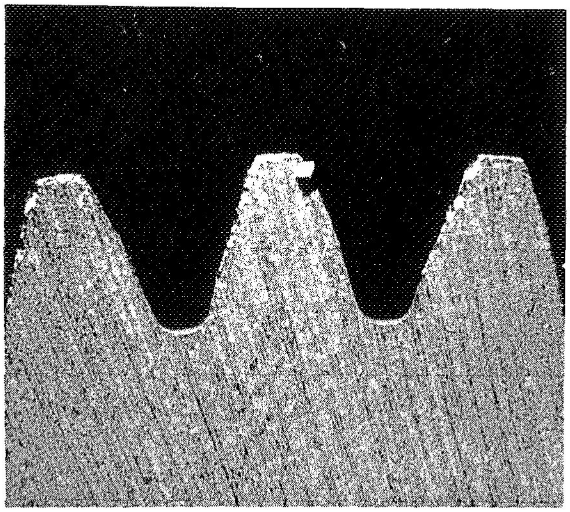
Material Combination	Test Duration hr	Test Number	Maximum Wear Percent	Material With Maximum Wear
II vs V	720	32	6.4	V
II vs VI	720	53	2.0	Same
II vs I	502	43	11.0	II
V vs VI	502	65	11.0	VI
IV vs VI	307	66	9.0	VI
V vs VII	120	31	17.0	VII
II vs VII	120	33	11.0	VII
VI vs III	61	55	---	Teeth sheared off VI
IV vs VII	21	22	14	IV
II vs IV	21	34	100	IV
II vs III	8.7	44	19	III
III vs VIII	5.5	56	19.0	VIII
IV vs VIII	4	21	16.5	VIII



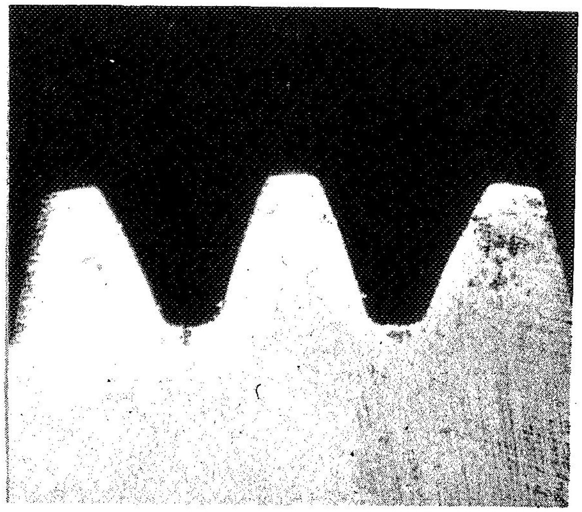
5-I



117-II



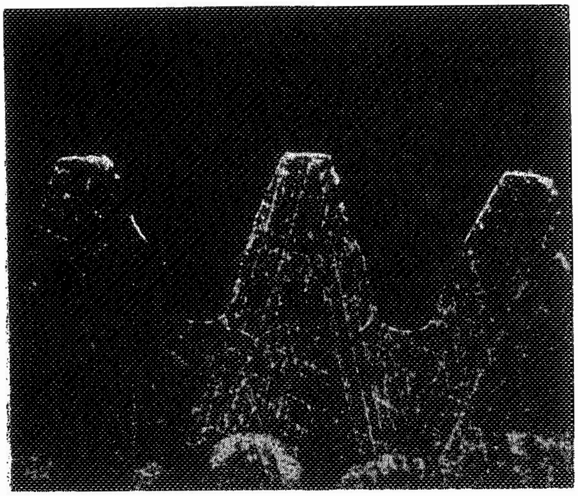
20-II



105-I

Environment - Vacuum
Duration - 133 hr

Figure 14 Test 39, Nitrided Nitralloy and Carburized C1020 Steel



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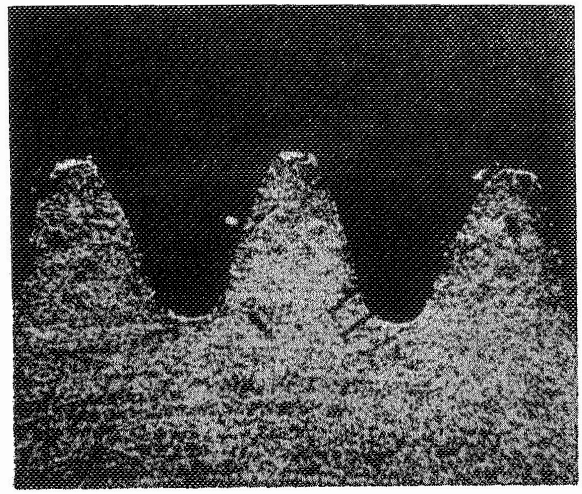
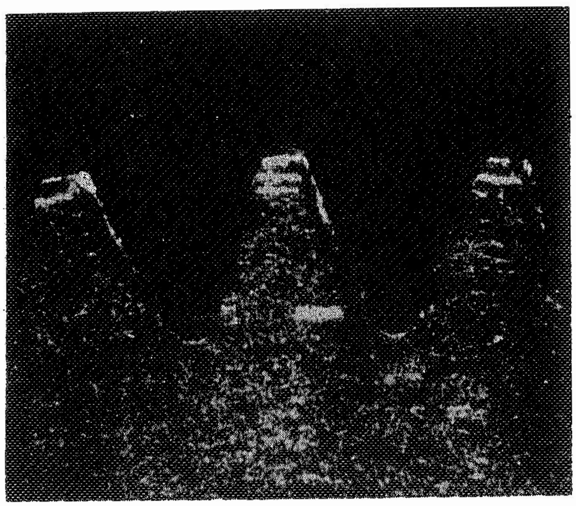
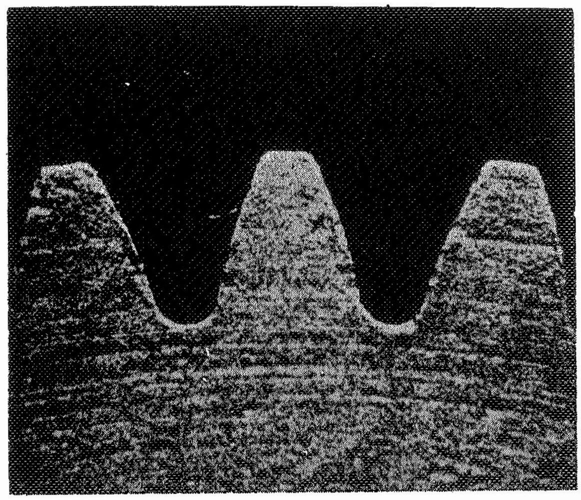


112-VII



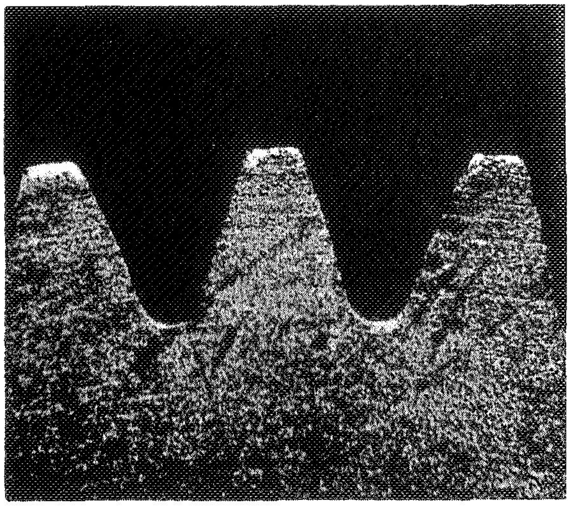
12-VII



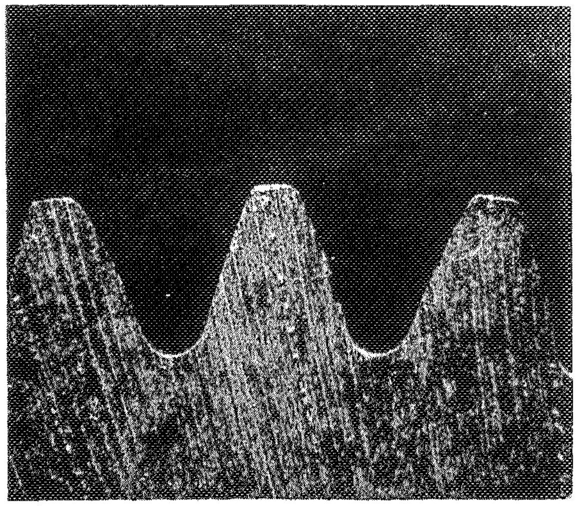
116-II

Environment - Vacuum
Duration - 720 hr

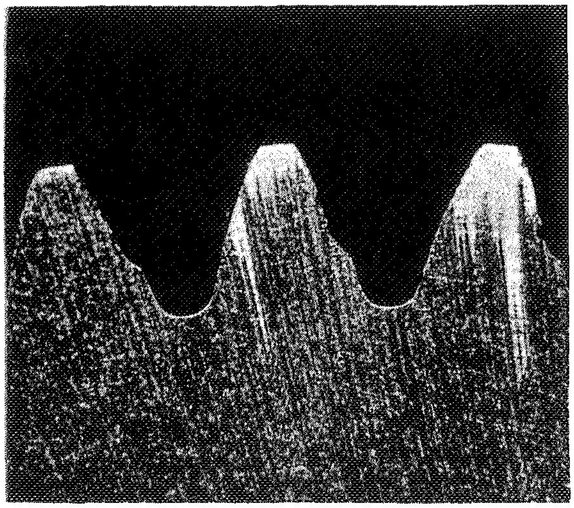
Figure 15 Test 38, Nitrided Nitralloy and C1085 Steel



7-VII



107-V



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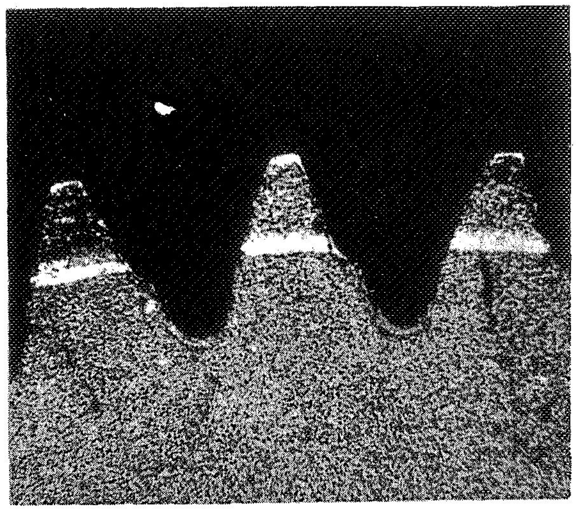
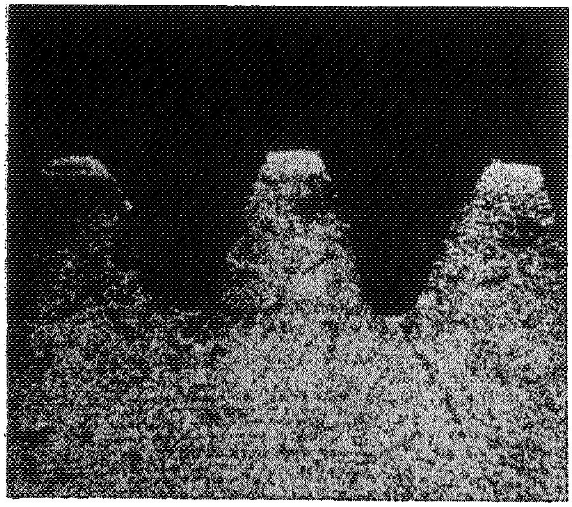


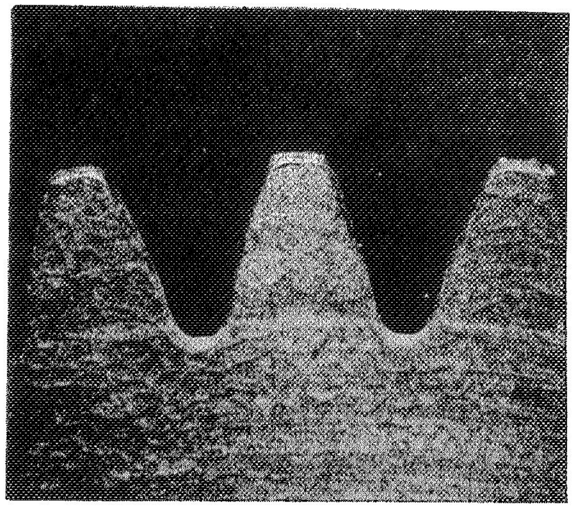
107-VII

Environment - Laboratory Ambient
Duration - 120 hr

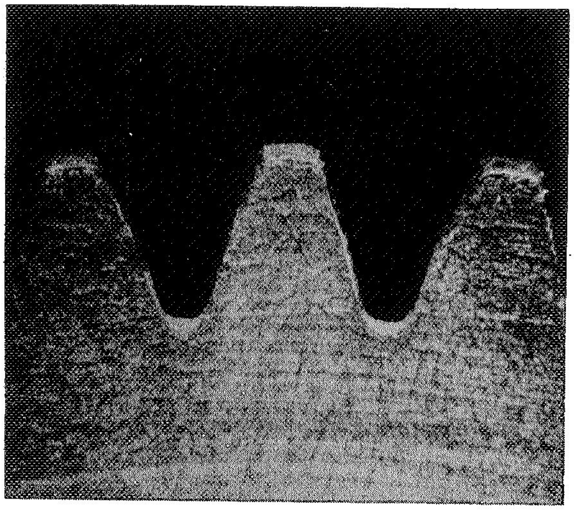
Figure 16 Test 31, 440C Stainless Steel and C1085 Steel



2-VII



111-IV



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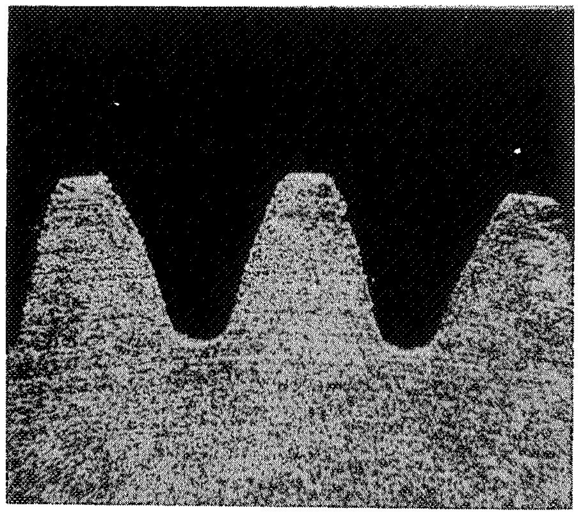
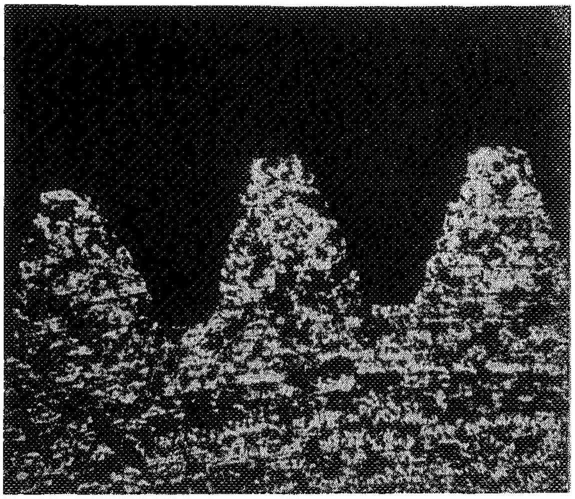


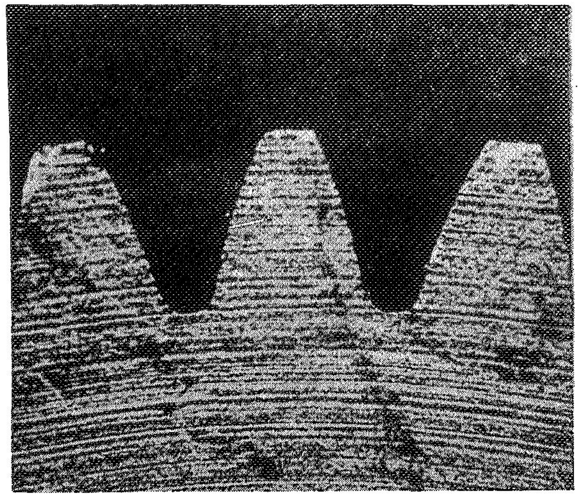
102-VII

Environment - Vacuum
Duration - 720 hr

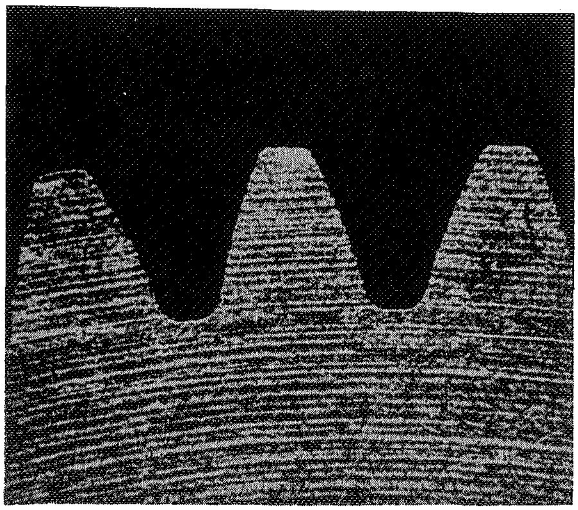
Figure 17 Test 26, Martin Hard Coated Aluminum and C1085 Steel



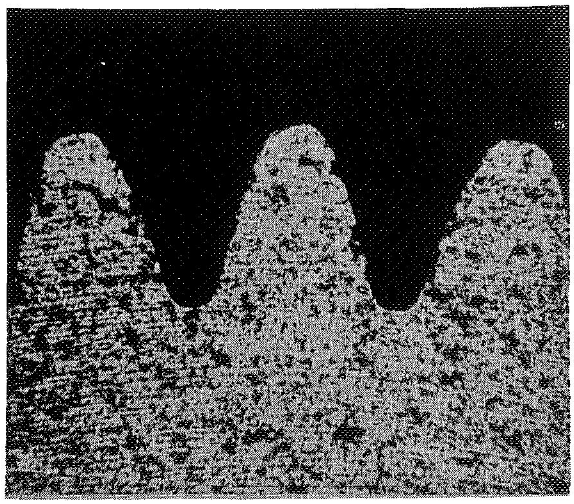
6-VI



126-II



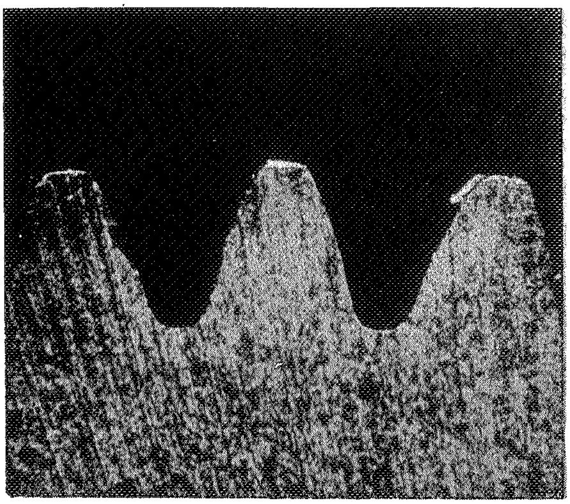
29-II



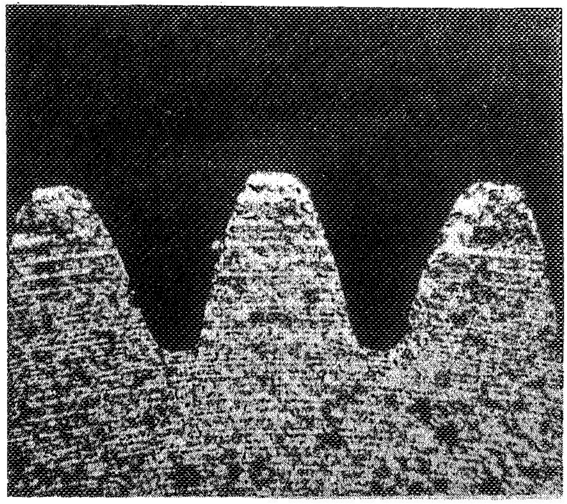
106-VI

Environment - Laboratory Ambient
Duration - 720 hr

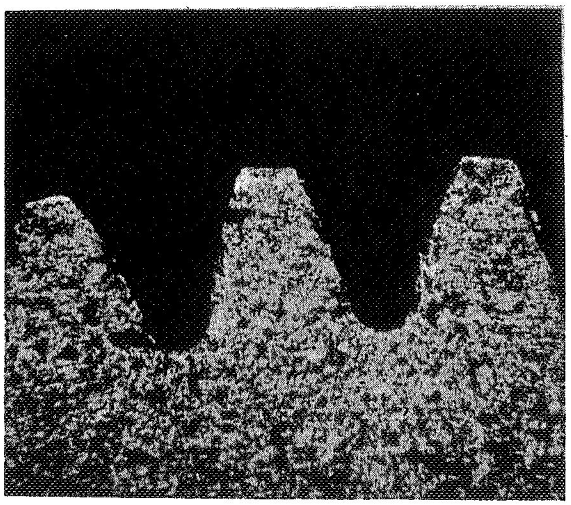
Figure 18 Test 53, Nitrided Nitralloy and Phosphor Bronze



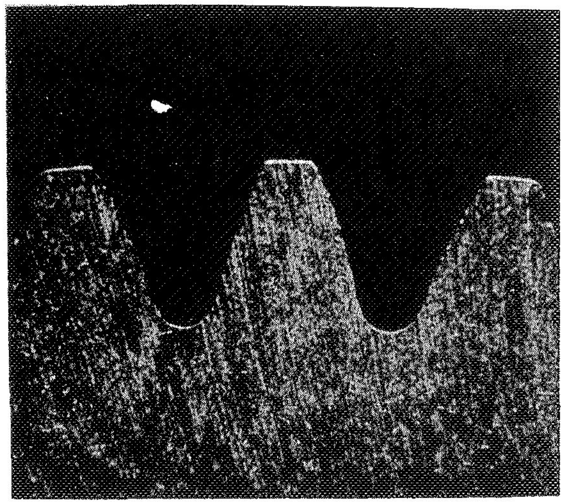
11-V



113-VI



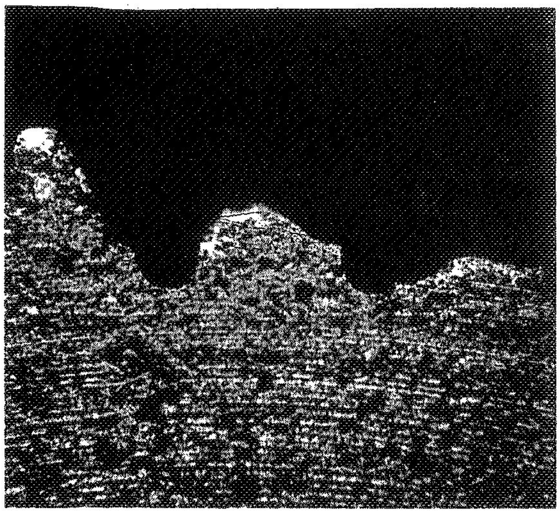
13-VI



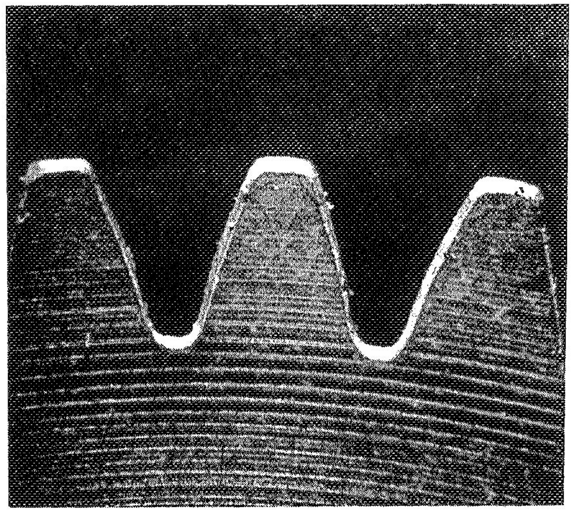
111-V

Environment - Laboratory Ambient
Duration - 502 hr

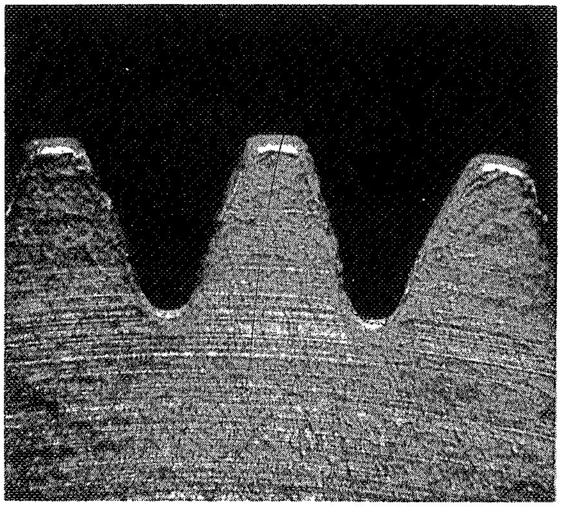
Figure 19 Test 65, Phosphor Bronze and 400C Stainless Steel



9-VI



119-IV



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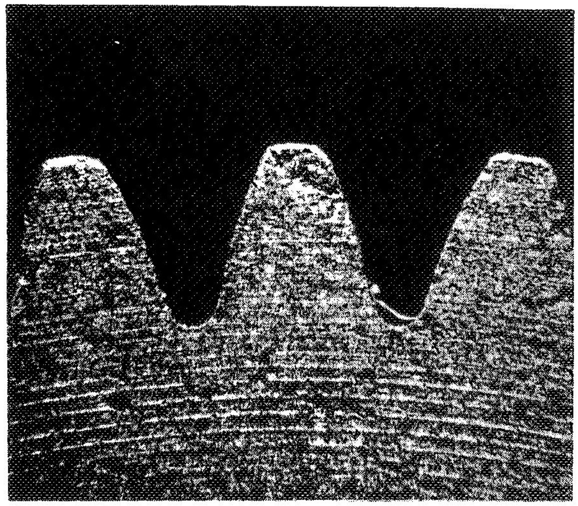
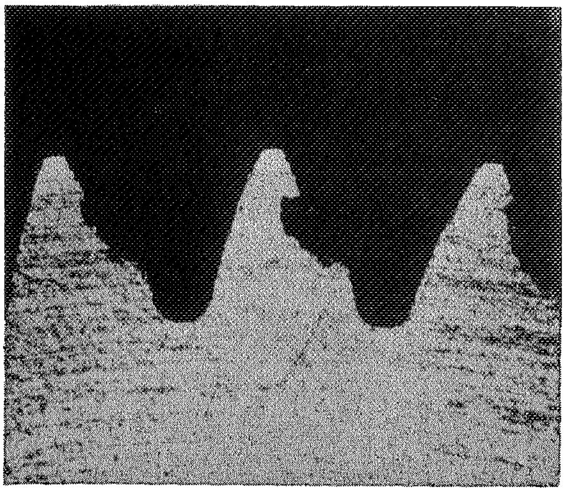


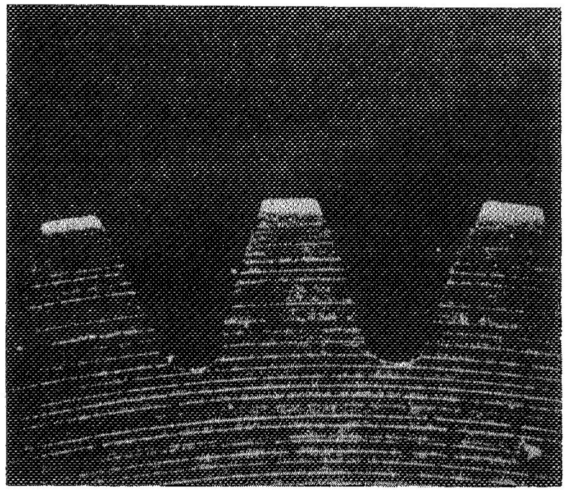
109-VI

Environment - Vacuum
Duration - 29 hr

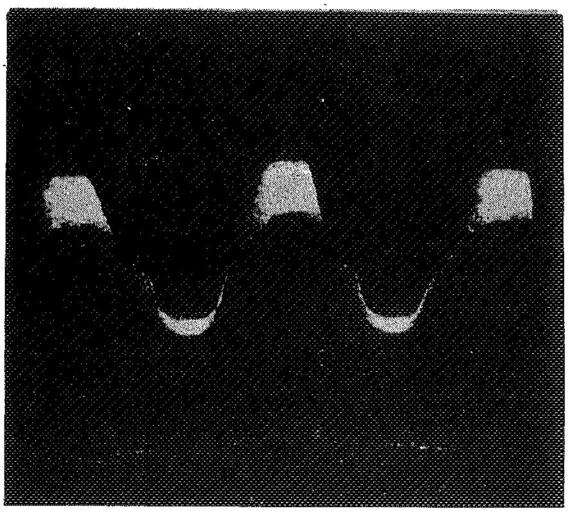
Figure 20 Test 61, Martin Hard Coated Aluminum and Phosphor Bronze



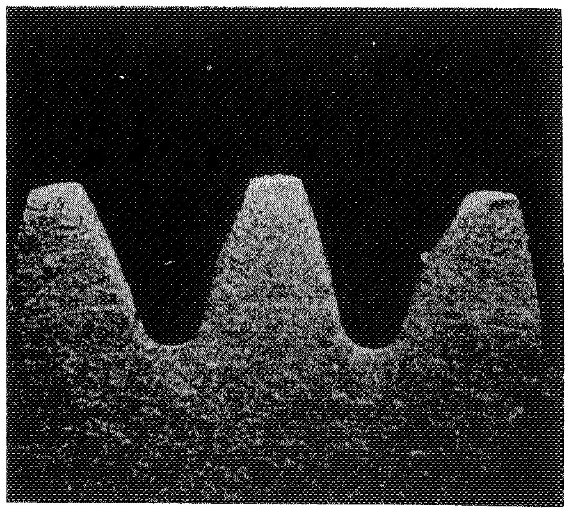
7-VIII



112-III



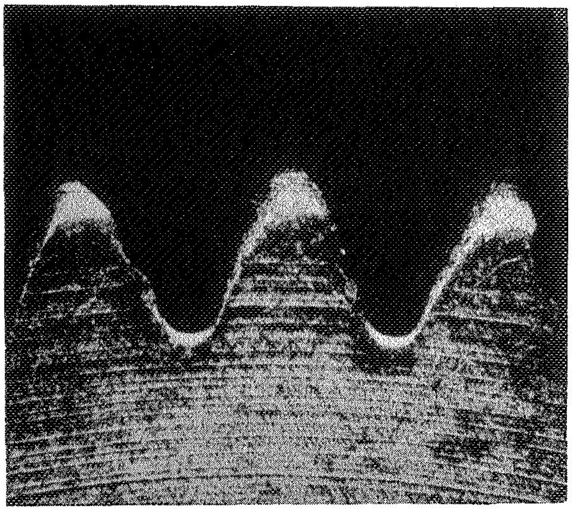
12-III



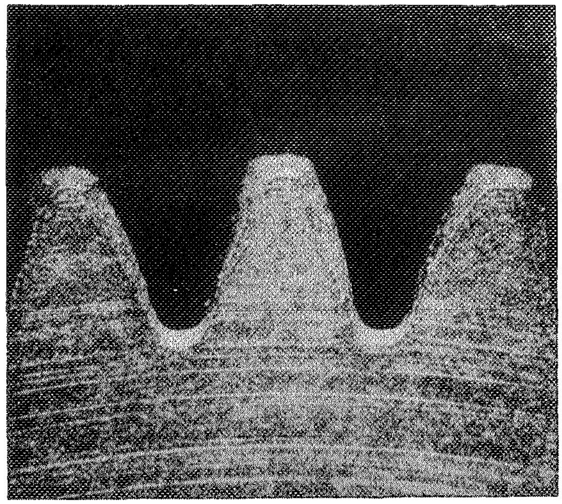
107-VIII

Environment - Laboratory Ambient
Duration - 5.5 hr

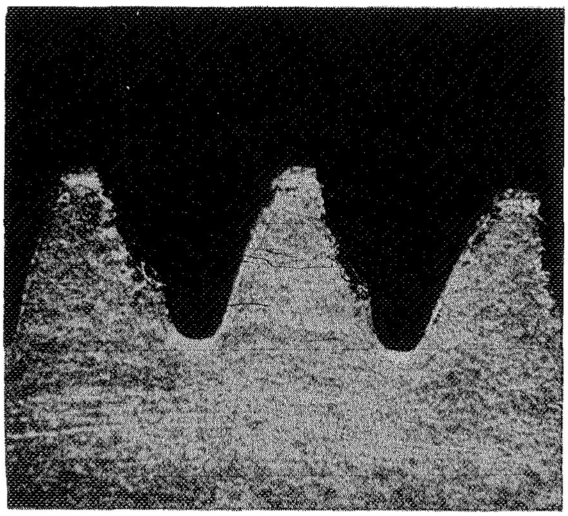
Figure 21 Test 56, Beryllium Copper and Light Anodized Aluminum



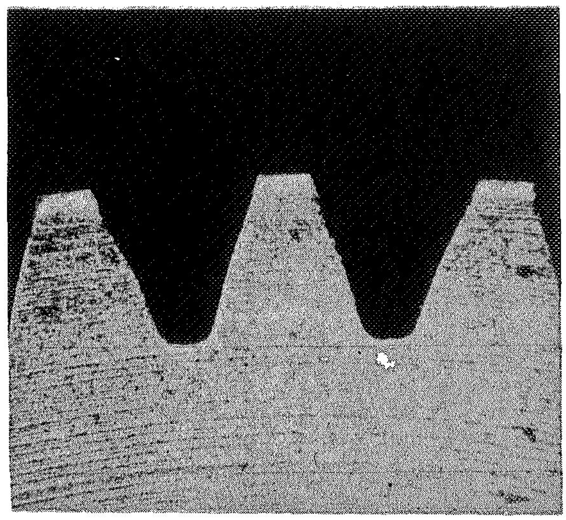
10-III



118-IV



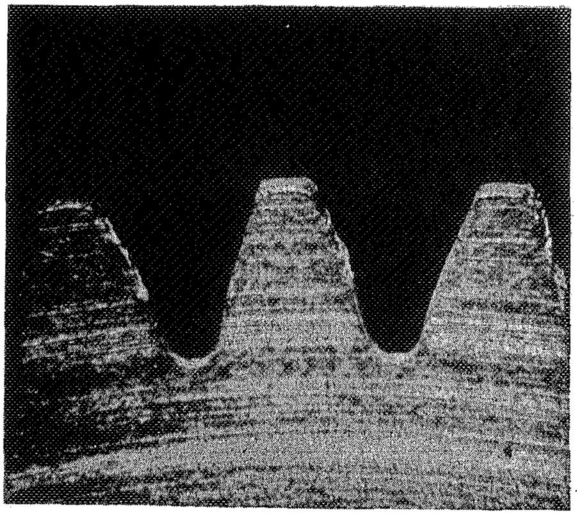
18-IV

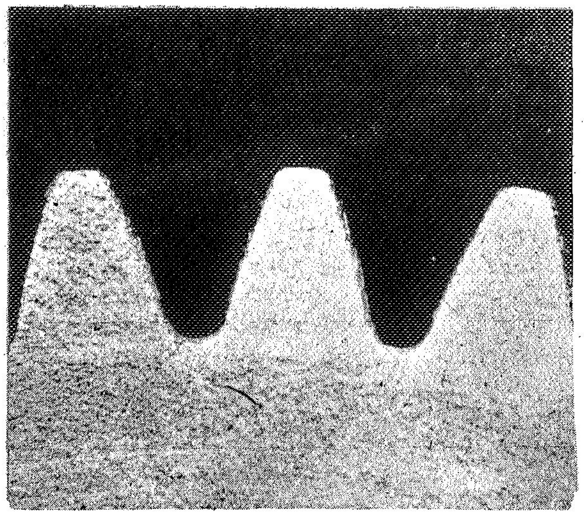


110-III

Environment - Laboratory Ambient
Duration - 19 hr

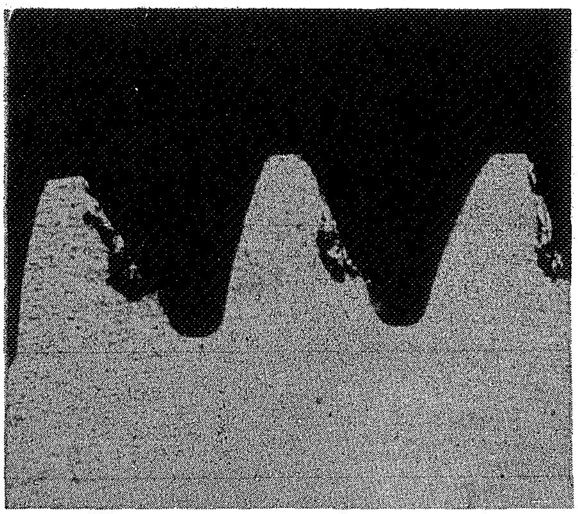
Figure 22 Test 44, Beryllium Copper and Martin Hard Coated Aluminum

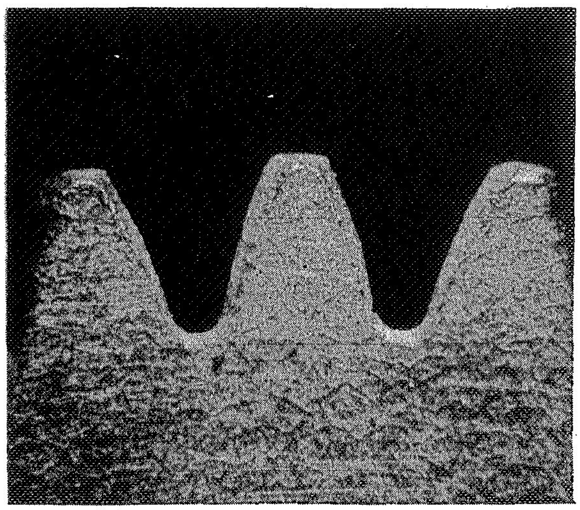




9-IV

105-VIII



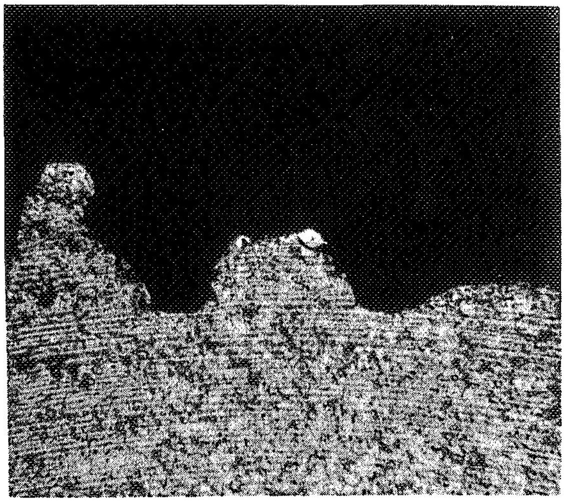


5-VIII

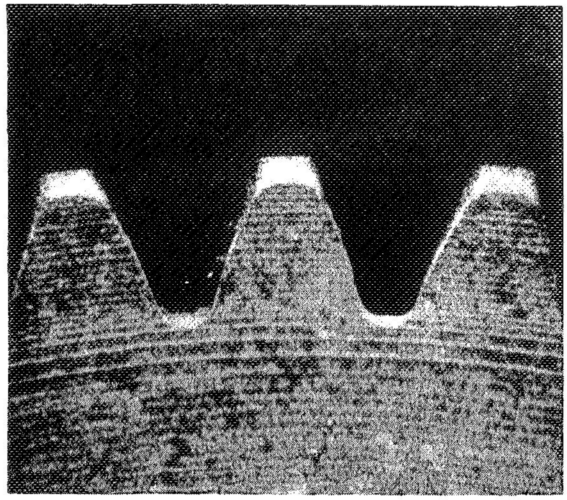
109-IV

Environment - Vacuum
Duration - 167 hr

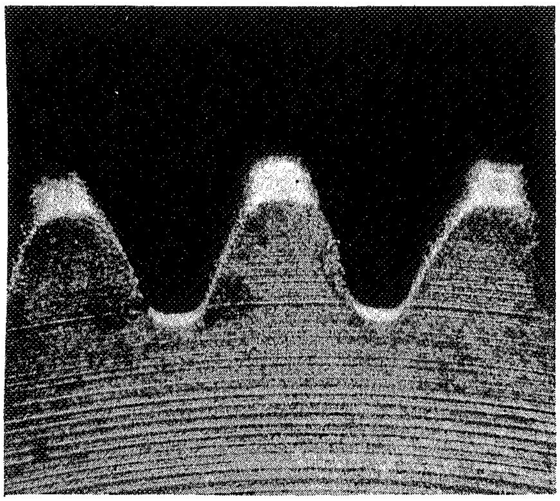
Figure 23 Test 24, Martin Hard Coated Aluminum and Light Anodized Aluminum



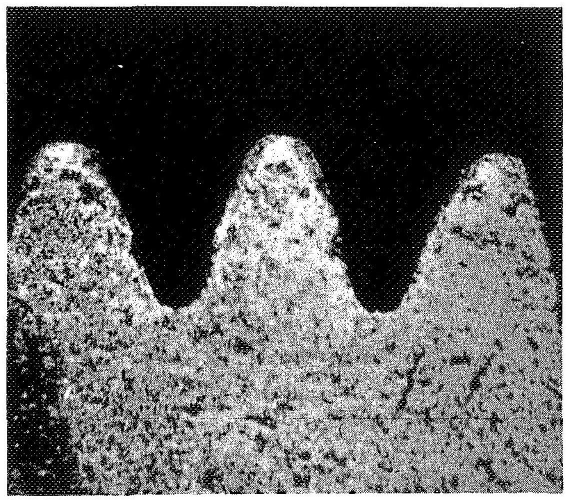
7-VI



111-III



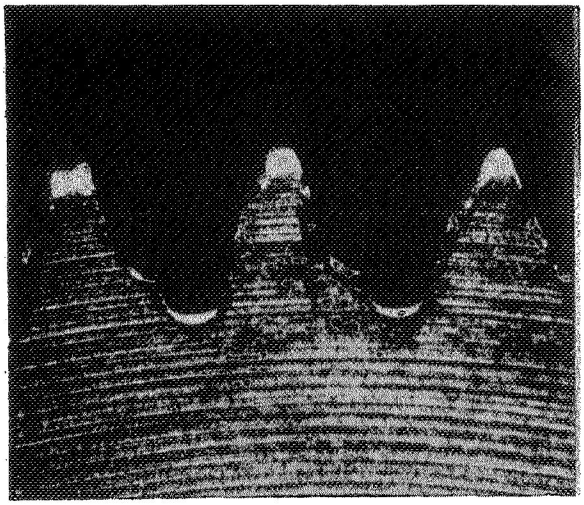
11-III



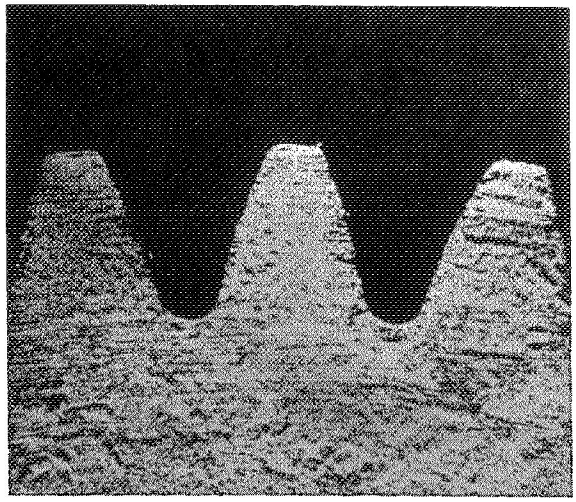
107-VI

Environment - Laboratory Ambient
Duration - 61 hr

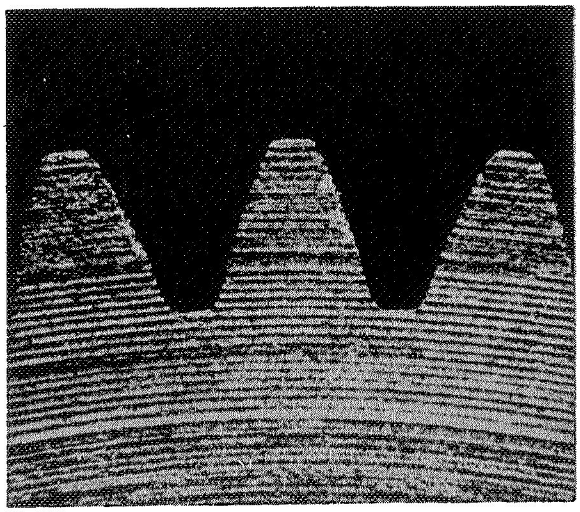
Figure 24 Test 55, Beryllium Copper and Phosphor Bronze



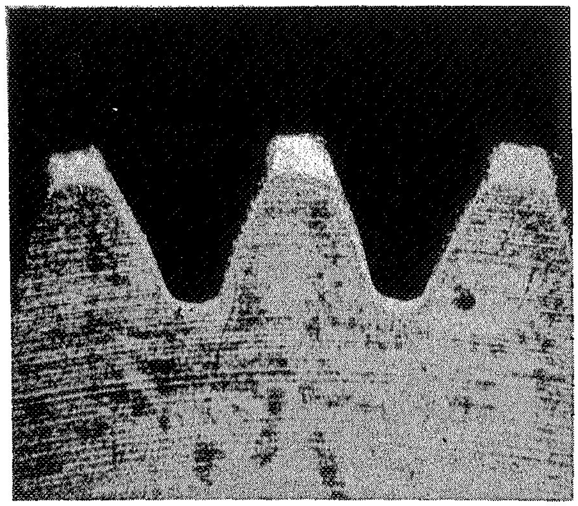
3-III



120-II



23-II



103-III

Environment - Vacuum
Duration - 24 hr

Figure 25 Test 42, Beryllium Copper and Nitrided Nitralloy

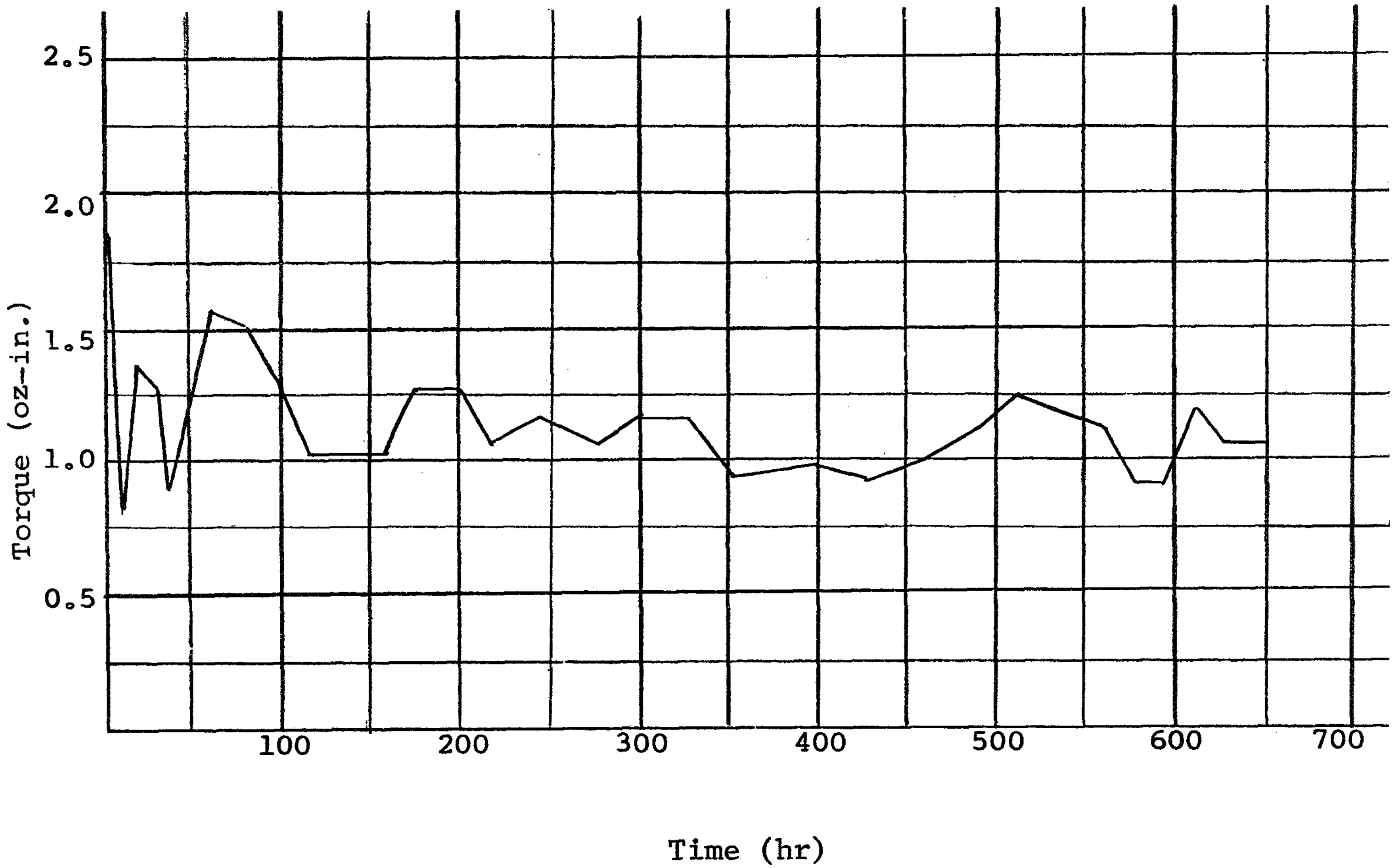


Figure 26 Torque Required to Rotate Gears as a Function of Time at 1800 rpm, 10 oz-in. Torque Load (Nitrided Nitralloy vs Nitrided Nitralloy)

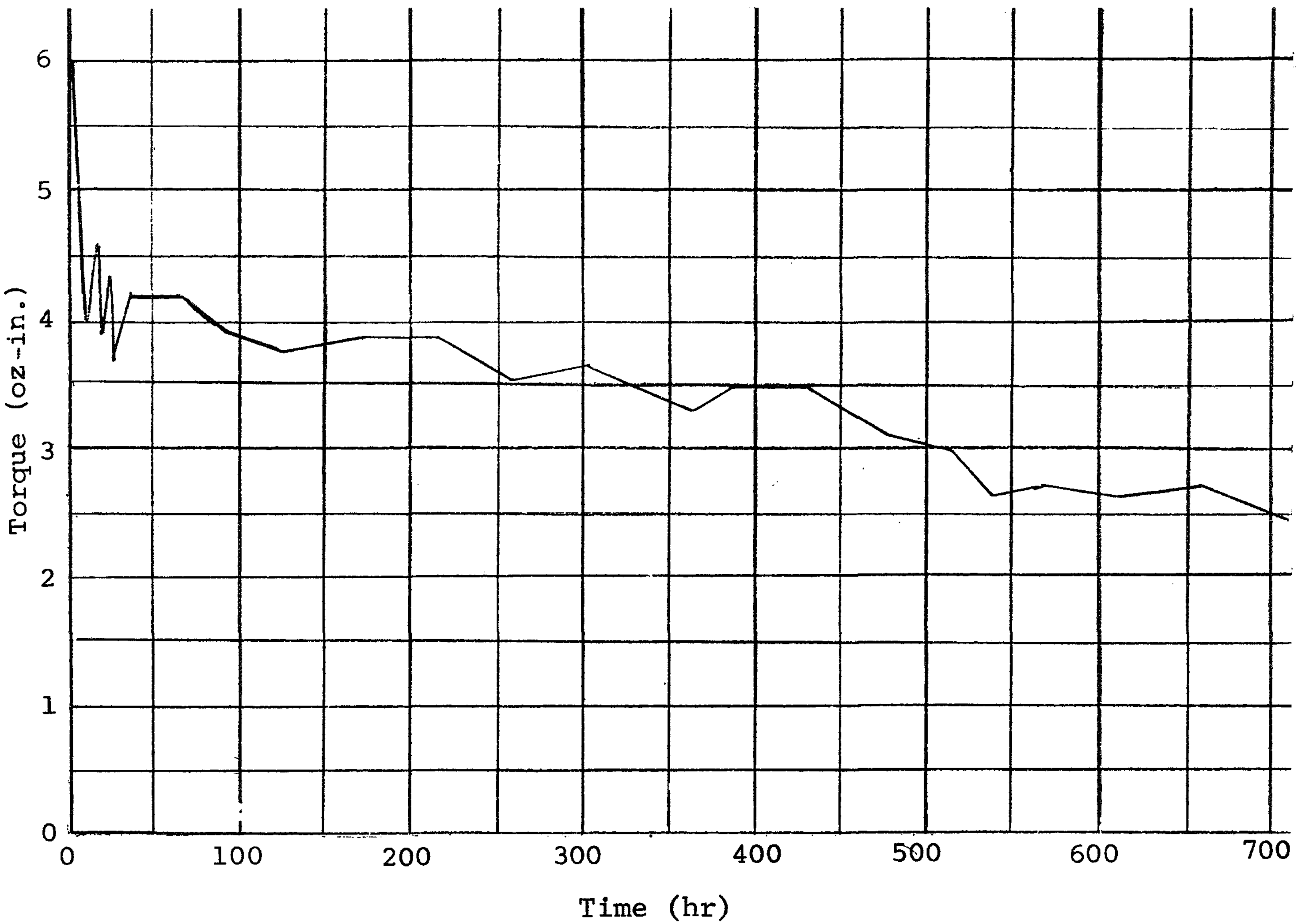


Figure 27 Torque Required to Rotate Gears as a Function of Time at 1800 rpm, 20 oz-in. Torque Load (Nitrided Nitralloy vs 440C Stainless Steel, Test 20)

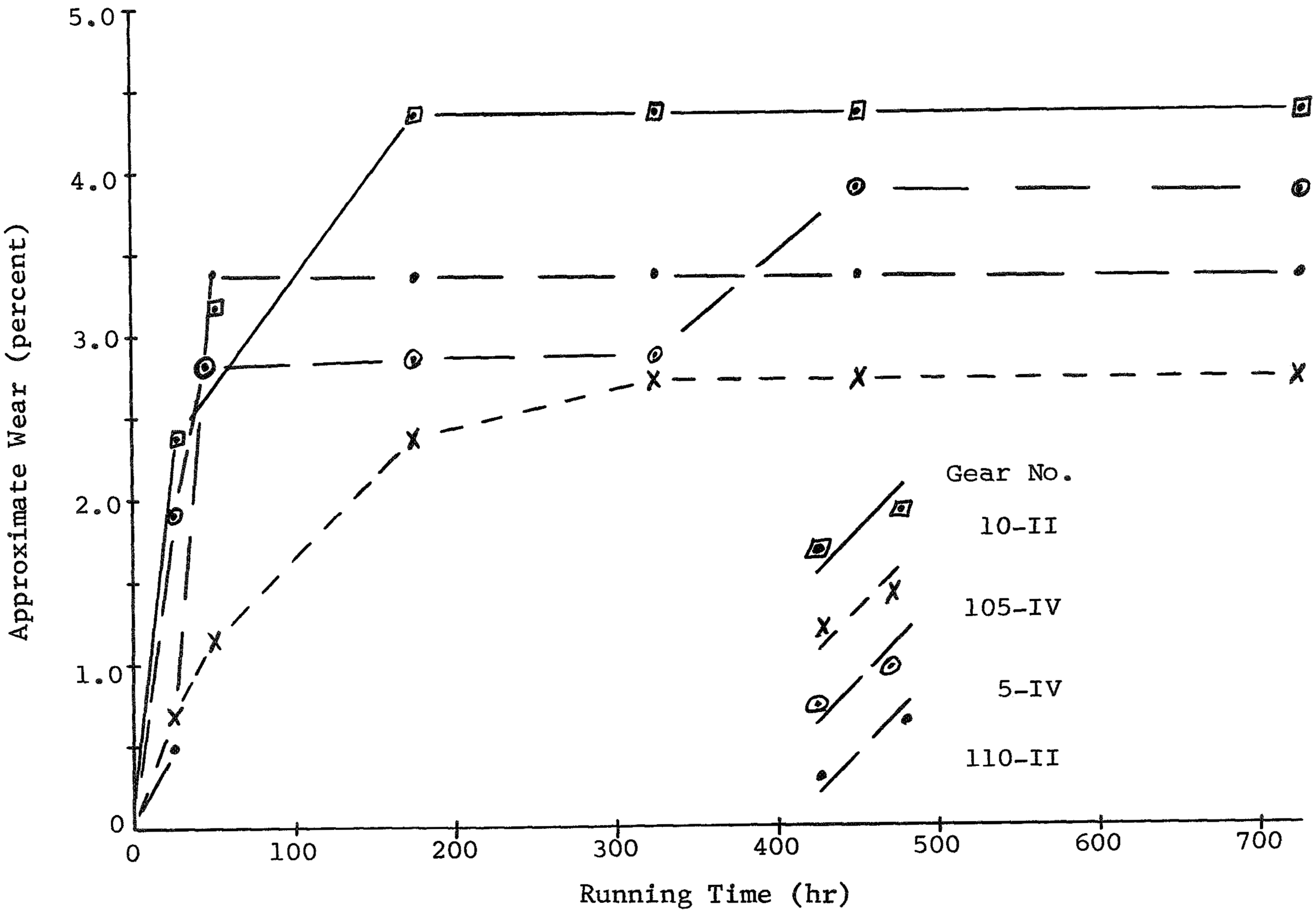


Figure 28 Wear Rate of Nitrided Nitralloy Mated with Martin Hard Coated Aluminum, Vacuum Test 15

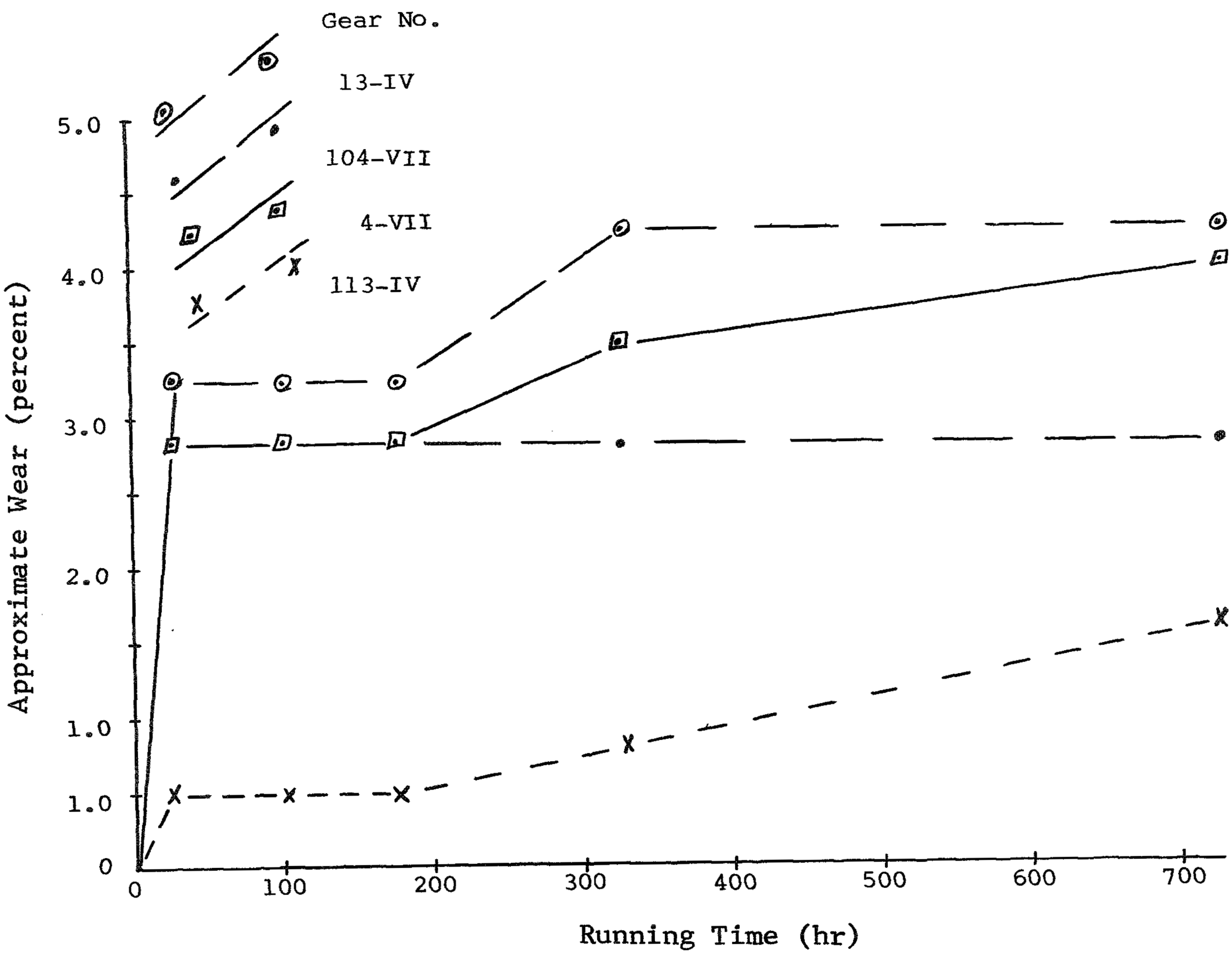


Figure 29 Wear Rate of Martin Hard Coated Aluminum Mated with C1085 Steel Silver Plated, with MoS_2, Vacuum Test 28

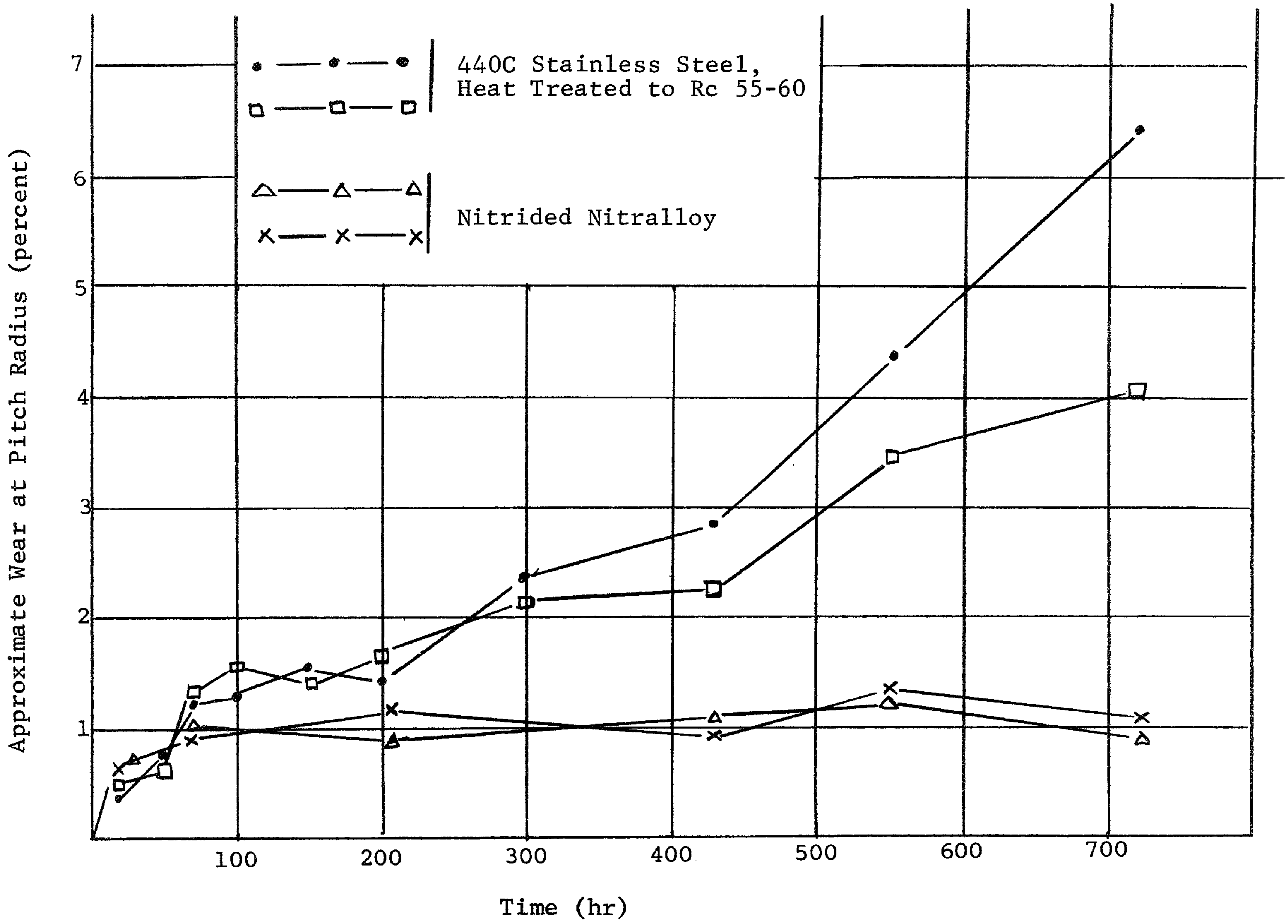


Figure 30 Wear Rate of Gears in Laboratory Atmosphere, Test 32

Table VII

SUMMARY OF TEST RESULTS ON UNLUBRICATED INVOLUTE GEARS (NITRIDED NITRALLOY VS 440C STAINLESS STEEL)

Environment	Test Number	Speed rpm	Torque oz-in.	Running Time hr	Percent Wear min	Percent Wear max
Vacuum	78 - 80	900	10	90 - 720	0	5.8
	91,92,96	900	20	274 - 498	0.6	6.3
	90,94,95	900	30	274 - 386	1.0	27.6
	86 - 88	1800	10	406 - 614	1.2	28.6
	82 - 84	1800	30	101 - 274	0.6	39
Laboratory Ambient	81	900	10	720	0	3
	93	900	20	945	0.6	2.0
	97	900	30	208	2	7
	89	1800	10	720	0	0
	85	1800	30	180	1	3

Table VIII

RESULTS OF VACUUM TESTS ON THIRD SET OF UNLUBRICATED INVOLUTE GEARS

NITRIDED NITRALLOY VS THROUGH-HARDENED 440C STAINLESS STEEL

Test Number	Speed rpm	Torque oz-in.	Percent Wear	Gear With Maximum Wear		Gear With Minimum Wear		Test Duration hr
				Dimension in.	Material	Dimension in.	Material	
106	1800	30	15	0.125	V	0.1875	V	650
107	900	20	5.5	0.1875	II	0.125	V	578
108	900	30	13.5	0.125	V	0.1875	II	578
109	1800	10	6.5	0.1875	II	0.1875	II	650

Table IX

RESULTS OF VACUUM TEST ON LUBRICATED INVOLUTE GEARS
(NITRIDED NITRALLOY VS MARTIN HARD COATED 7075T6 ALUMINUM
ALLOY, SHAFT SPEED 1800 RPM, TORQUE LOAD 20 OZ-IN.)

Test Number	Lubricant*	Environment	Test Period hr	Gear With Maximum Wear			Gear With Minimum Wear			Remarks
				Percent Wear	Dimension in.	Material	Percent Wear	Dimension in.	Material	
111	A	PL 5×10^{-8}	364	15.3	0.125	IV	<0.5	0.1875	II	IV corners chipped
112	B	PL 5×10^{-8}	364	4.3	0.125	II	<0.5	0.1875	IV	IV corners chipped
113	A	PL 5×10^{-8}	312	2.3	0.1875	II	0.9	---	IV	IV corners chipped
114	A	Lab Ambient	216	5.8	0.125	IV	1.5	0.1875	IV	IV
115	B	$p < 6 \times 10^{-8}$	340	6.1	0.125	IV	2.4	0.1875	II	IV edge rounded
116	B	$p < 6 \times 10^{-8}$	340	1.1	0.1875	II	<0.5	0.1875	II	Edge rounded
117	B	$p < 6 \times 10^{-8}$	238	2.3	0.1875	II	<0.5	0.125	IV	IV edge chipped lube flaked
118	B	Lab Ambient	119	6.1	0.125	IV	<0.5	0.1875	II	IV badly chipped
119	C	$p < 6 \times 10^{-8}$	340	3.2	0.125	II	1.2	0.125	IV	Lube flaked
120	C	$p < 7 \times 10^{-8}$	66	6.7	0.125	IV	<0.5	0.1875	IV	Rounded corners
121	C	Lab Ambient	216	5.3	0.125	IV	<0.5	0.1875	IV	0.125 in. IV, one tooth badly chipped

*A - Burnished MoS_2
B - Electrofilm
C - Ultrathin (Dow Corning Co.)

Under laboratory ambient conditions, wear can be monitored directly and efficiently; however, wear is less observable in vacuum tests. Tests 114 and 121 showed less than 1 percent wear after 195 hr. This increased to nearly 6 percent during the next 24 hr. The wear followed a describable pattern. During the first day of testing a sprinkling of debris was deposited on the support stand of the test fixture. This of course was carefully observed and it was noted that there was no (or very little) increase as the test progressed. The day before the test was ended there was very little debris. The next day the support stand of the test fixture and the bearing support were covered with debris, some thrown off the gears by the centrifugal forces.

After observing this catastrophic failure we watched carefully for debris deposition in the vacuum chamber and when a sudden debris increase occurred the testing was halted. Three gears out of the complete set of 44 show less than 4 percent wear. However, information can be gleaned from sectioning and studying these gears along with the others. Also it is unlikely that they would have lasted much longer. Figures 31 through 34 present sections through the teeth of gears which were used in these tests.

Cycloidal Gears

A detailed tabulation of the results generated during the cycloidal profile gear evaluations is shown in Table X. This includes the results from six vacuum tests and two atmospheric tests. During this phase of the program only carburized C1020, nitrided nitralloy, and 440C stainless steel materials were evaluated. As the table shows, none of the cycloidal gears fabricated from these materials successfully completed the 720 hr test period. Based on a time reference the cycloidal gear tests were completed immediately after the first 224 involute gear tests.

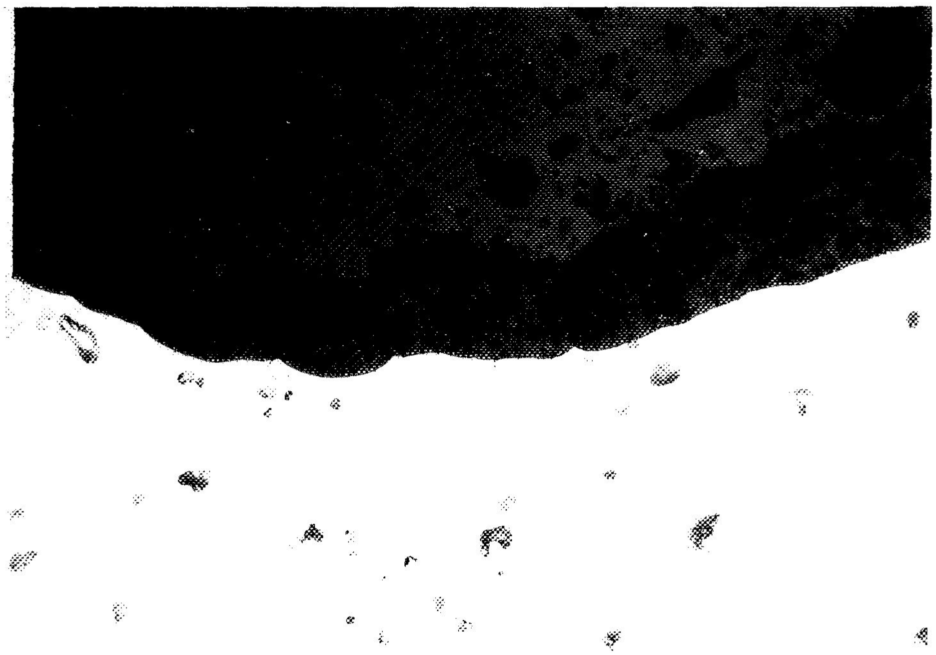


Figure 31 Metallograph of Martin Hard Coated Aluminum Gear from Laboratory Environment Test, 21 hr

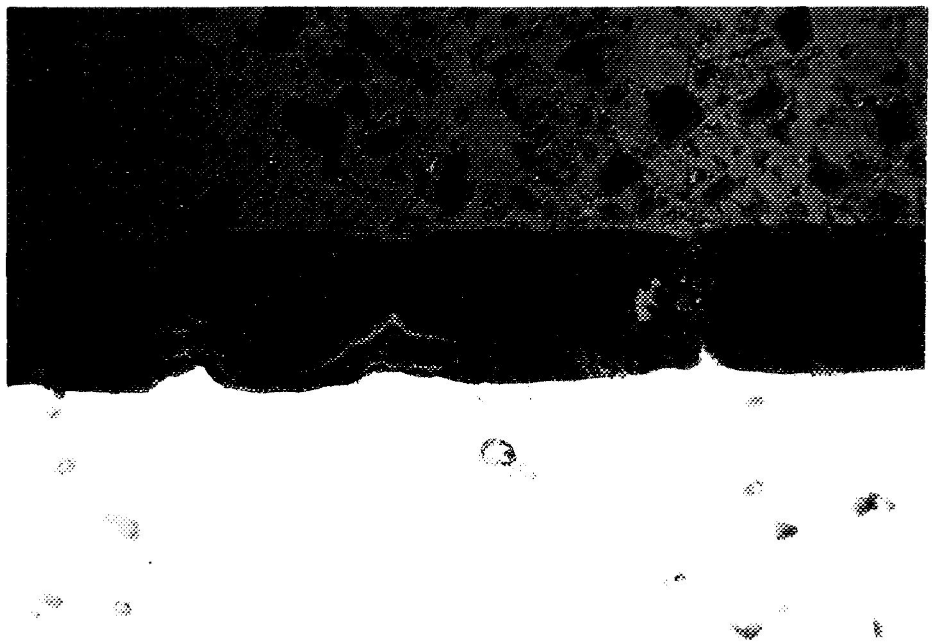


Figure 32 Metallograph of Martin Hard Coated Aluminum Gear from Vacuum Test, 720 hr

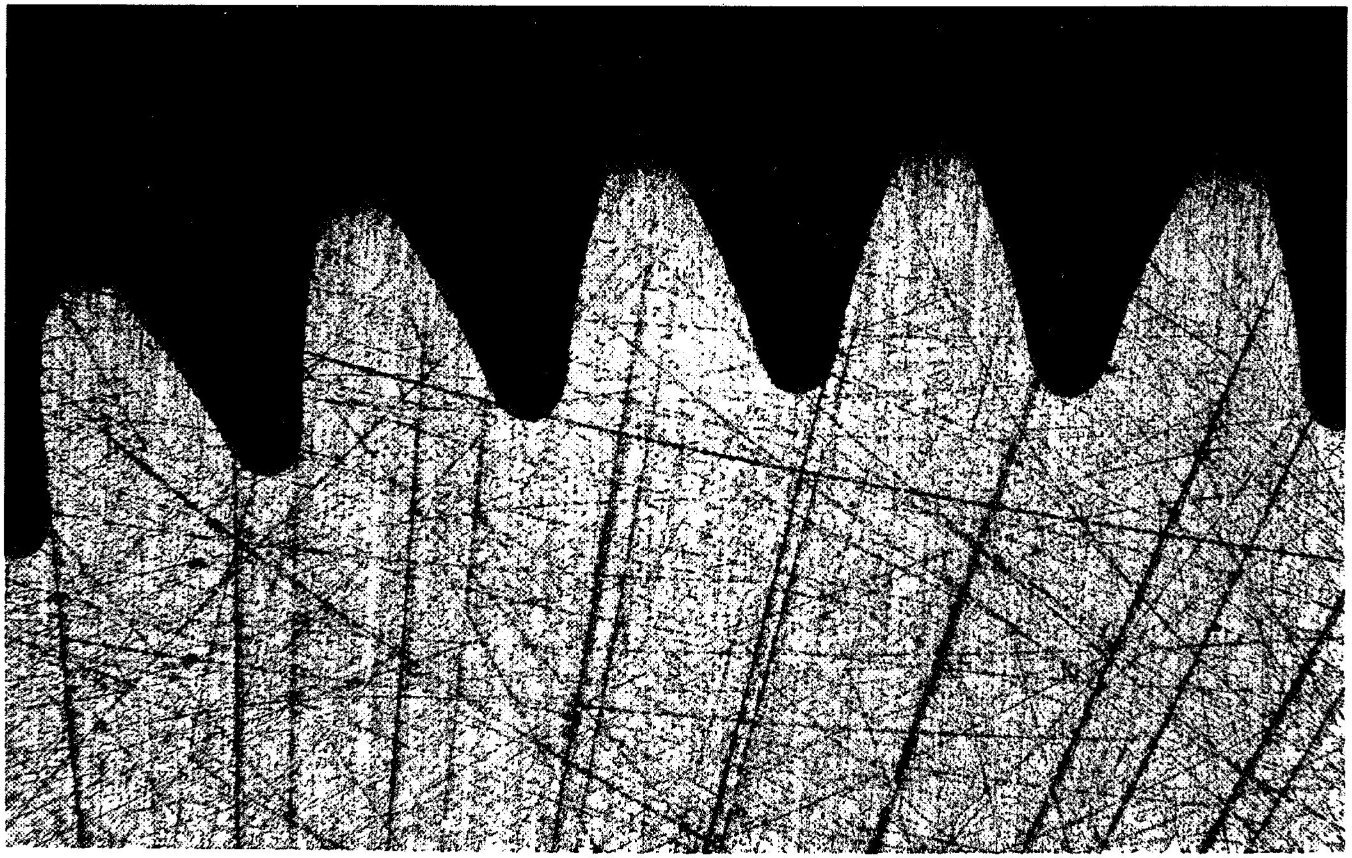


Figure 33 Test 106, 440C Stainless Steel Involute Gear

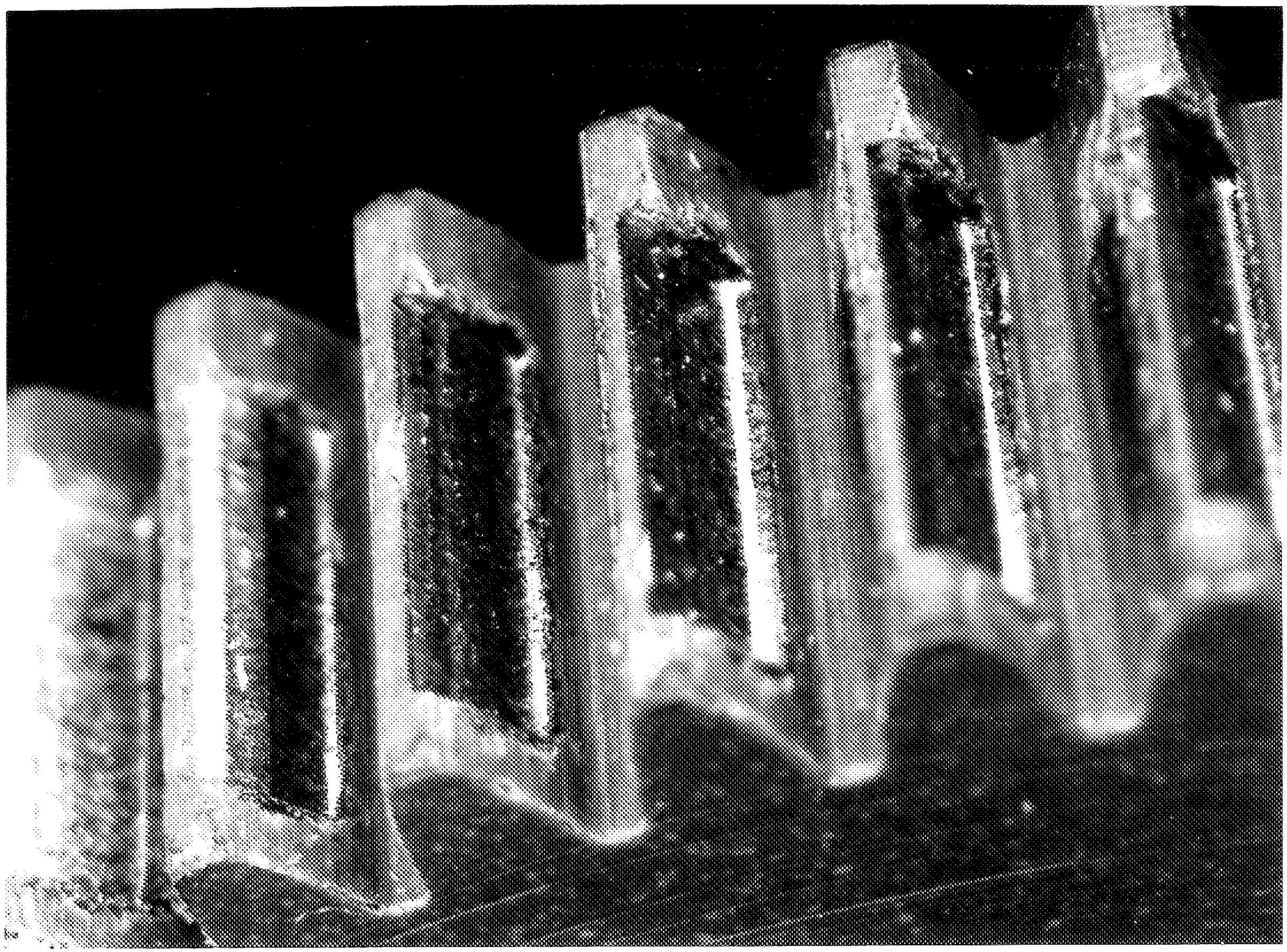


Figure 34 Test 113, Martin Hard Coated Aluminum, MoS_2 Lubricated, Involute Gear Test in Vacuum, 200 hr

Table X

CYCLOIDAL GEARS TESTED AT 1800 RPM

Test Number	Material Combination	Environment	Torque Load oz-in.	Test Duration hr	Percent Wear	
					min	max
69	II vs V	Lab Ambient	20	277	3.7	10.4
70	II vs V	Vacuum	10	600	2.5	9.2
71	II vs V	Vacuum	20	457	0.4	10.4
72	I vs V	Lab Ambient	20	Test terminated, insufficient backlash		
74	I vs V	Lab Ambient	20	15½	6.0	13.3
73	I vs V	Vacuum	20	600	0.9	3.0
75	I vs II	Vacuum	20	14	1.2	9.9

Apparatus Reliability

The established test period (720 hr) provided an excellent opportunity to evaluate the reliability of many components in a vacuum environment. Some observations made during the study are discussed.

Motors. - The 24 dc motors used to drive the four-square test fixtures are used for ~2500 hr without any problem. At that time the motors are returned to the manufacturer for replacement of the brushes and commutators. Figure 5 shows that the motors are located outside of the vacuum chamber. The rotational torque is coupled by magnetic flux through a nonmagnetic membrane.

SR4 bearings with Salox M retainers. - Very good wear characteristics are obtained with the SR4 precision quality bearings using Salox M retainers. Initially, some doubt was expressed as to the feasibility of using such bearings for 720 hr in a vacuum environment. However, not only can these bearings be used for 720 hr, but several emerge in very good condition. Some are even reusable for subsequent tests. At the end of 1440 hr of accumulated usage, these bearings fail in a manner very similar to other bearings which are unusuable after only 720 hr.

A relationship was found between the amount of wear which occurs on the test gears and the damage sustained by the bearings. After the bearings are used for 720 hr they are cleaned by blowing air through the bearing while allowing it to rotate. This procedure apparently removes loose wear particles from the bearings and in cases where the bearing movement has not degraded significantly, the bearings are reusable. The condition of the bearings is usually quite good where only a few percent wear occurs on the test gears. This relation to gear wear holds even when the bearings have dust shields. This is to be expected since with increasing wear vibration increases; the bearings are thus under increasing vibrational loading.

It should be noted that these dust shields have clearances of 0.001 to 0.002 in. and, therefore, wear particles can find their way into the races. For this reason, it may be beneficial to use bearings which have a teflon seal to prevent this problem.

The type of failure which occurs in the bearings is a combination of surface fatigue, and abrasive and adhesive wear which result in pitting of the races and balls. This type of wear, particularly in vacuum, results when inadequate lubrication is present and thus metal to metal contact predominates. The lack of, or the small amount of lubrication, is evidenced by the only slightly worn condition of the Salox M retainers. In addition, many of the used balls have a characteristic bronze color which indicates adhesion in the absence of sufficient lubrication.

<u>Four-square test fixtures</u>. - The four-square test apparatus shown in Figure 4 performs well during tests at 10 and 20 oz-in. torque loads; however, problems arise when the torque load is increased to 30 oz-in. We found it necessary to replace all the slotted head stainless steel screws in the gear clamps on the test fixtures with hardened socket head cap screws to prevent rotation of the gears.

MODES OF GEAR SURFACE DETERIORATION

The forms of gear deterioration which predominate with each combination of materials are interpreted by simultaneously examining the physical characteristics of the materials and the test environment. These modes may be grouped as follows:

Wear
Surface Fatigue
Plastic Flow
Tooth Breakage

These are primarily controlled by the gear geometry, gear material and lubrication. The primary variable in this study is gear material, the effect of lubricants is secondary. A brief discussion of the cycloidal and involute gear profiles will help to relate the influence of gear geometry on the test results.

Although a series of cycloidal gears is evaluated, the major effort is expended using involute gears. The problem of fabricating precision cycloidal gears and the sensitivity of these gears to changes in center distance severely limits their use for instrument size spur gears. The use of the involute gears is thus dictated even though it is recognized that this profile has one very important limitation. Namely, since external gears using involute profiles have convex active surfaces, the relative curvatures in the contact zone between these gears is quite severe. For this reason the contact stresses are high. The only method of reducing these stresses for a particular gear diameter is to increase the pressure angle, since the relative curvature of the teeth is independent of the pitch of the teeth at any point of contact.

An alternative gear profile which may hold promise for vacuum application is the modified cycloidal profile known as the circular arc of the Wildhaber-Novikov form. Although this profile is more sensitive to center distance variation than the involute form, it is more tolerant than the normal cycloidal profile.

The circular arc consists of a concave surface mating with a convex surface and the contact stresses are consequently considerably reduced. Tests with involute profile and Wildhaber-Novikov gears indicate that at the same loads the Novikov outlasts the involute by 3 to 5. Such results merit serious consideration for gears required for use over long periods of time. These gears are essentially circular and must be used as helic gears in order to satisfy the fundamental gear law. Therefore, they are not spur gears and should probably be considered only for long-life applications where the contact stresses will significantly reduce gear life.

Wear

Wear is a phenomenon which is characterized primarily by the deterioration of material surfaces when exposed to dynamic mechanical and/or chemical environments. The deterioration can be manifested by loss of material and a change in the characteristics of the surfaces in contact in such a manner that performance is degraded. In more severe cases, the deterioration leads to scoring or scuffing of the contacting surfaces. A detailed discussion of this phenomenon as well as descriptions of existing theories are presented in Appendix D.

Surface Fatigue

Areas near the points of contact of rolling spherical or cylindrical surfaces are subject to Hertz stress concentrations and cyclic contact can therefore lead to high values of local cyclic stress. This leads to a fatigue failure resulting in the removal of a relatively large chip or in spalling or pitting of the surface. This type of wear is catastrophic and generally causes a part to become inoperable quite suddenly. The magnitude of the contact stress and the number of cycles to failure are related in this kind of fatigue in very much the same way as stresses and cycles in a traditional S-N curve. The exact character of the maximum stress depends not only upon the normal load but also the properties and dimensions of the rolling surfaces and upon the ratio of rolling to sliding of the surfaces. Lubricants are effectual in reducing wear by preventing adhesion, and reducing the coefficient of friction so that the Hertizian contact stresses are insufficient.

Plastic Flow

Plastic flow occurs on gear surfaces as a result of high stresses. This type of gear failure is normally indicated by finned material overhanging the tips of the teeth and is more predominate with soft and medium hard materials which have relatively low yield strengths. To eliminate plastic flow the load (total or contact) must be reduced or the load bearing capability of the tooth must be increased by material selection.

Tooth Breakage

Tooth breakage is caused by overload, shock, or common fatigue. Either a complete tooth or a portion thereof may be broken off.

TEST RESULTS

The test results generated on this program are not intended to provide solutions to the wide variety of problems associated with the prediction of wear in vacuum, however, they do provide guidelines for the selection of material combinations for use in vacuum. In this respect they are significant when viewed and interpreted as a series of screening tests for a wide variety of materials.

Thus, it should be remembered that the level of contact stress imposed on the materials tested will have a much more marked influence on the rate of wear of some material combinations than on others. These differences result from the variation of fatigue characteristics of the individual materials as well as the wear resistance of the materials as pairs. The contact stress, in these tests, is $\sim$60,000 psi which is far below the fatigue strength of the harder materials, but is very near the limiting stress for a material such as Phosphor bronze with a 15 percent MoS_2 matrix.

Nitrided nitralloy vs 440C stainless steel. - This combination endures well in both laboratory and vacuum environments. In both series of tests the primary mode of surface degradation is by abrasive wear. The fact that both of these materials have very hard but chemically different surfaces may account for the good performance. In one case the surface is composed of dispersed fine nitrides and in the other dispersed fine carbides. The nitrided nitralloy surface hardness approaches 1200 knoop which is attributed to the state of fine dispersion of the nitride particles in the matrix rather than the inherent hardness of large nitride grains. The 440C stainless steel gear has a hardness of ~700 knoop plus a structure containing chromium carbides and intermediate carbon martensite.

This combination of materials may be usable for extended periods of time, since the nitrided nitralloy gears are slightly harder than the stainless ones and thus should wear at a slower rate. Since the stainless steel gears are completely hardened the rate of wear will not cause these gears to fail after a few thousandths wear has occurred. However, the life of such a combination could possibly be extended by using a deeper nitrided case, since this would allow additional wear at or near the maximum hardness of the nitrided case.

Nitrided nitralloy vs carburized C1020. - This combination does not successfully complete all tests in either vacuum or laboratory environments. The initial mode of failure in all cases appears to be excess wear. Test 16 successfully completed the 720 hr test, and both materials wore approximately the same amount. However, in all other tests the wear occurs more rapidly on the carburized C1020 gears. In the advanced stages of wear some evidence of plastic flow of the carburized gear occurs.

The useful life of gears made from this combination of materials could be extended by increasing the depth of both the carburized and nitrided cases. This seems to be very desirable in the carburized gears, since the substrate material is very soft and not capable of supporting the load, after only a very small percentage of wear has occurred.

Nitrided nitralloy vs C1085 silver plated and MoS_2. - This material combination provides some significant information about the operating parameters. Two of the three vacuum tests run 720 hr and the third set runs 393 hr; however, the set tested in the atmosphere runs only 120 hr. In all these tests the C1085 gears show evidence of plastic flow. These gears are heat treated to Rc50 prior to being plated.

Unfortunately the tensile strength of C1085 steel heat treated to Rc50 as a function of temperature is not available in the literature. However, published values of tensile strength as a function of temperature for C1030 heat treated to ~Rc33 (refs 1 and 2) should be quite similar to C1090 or probably lower.

Interpolation of these published values of tensile and yield strengths indicates that the temperature of the asperities on the wear surface probably exceeds 800°F before they flow plastically at a contact stress of 60 000 psi. Examination of the mating nitrided gears tested in vacuum indicates that the silver plate and MoS_2 coat the nitrided wear surface and should reduce the sliding friction during operation. However, in air the MoS_2 is much less effective, because of oxidation at the elevated operating temperature. This may account for apparent better operation of the combination in vacuum.

The nitrided case shows evidence of brinnelling and surface fatigue and subsequent material transfer after ~8 percent wear. This tends to substantiate the need for a deeper nitrided case.

Stainless steel 440C vs C1085 silver plated and MoS_2. - This combination does not complete the 720 hr test in either vacuum or laboratory environments, primarily because of plastic flow on the surface of all the C1085 gears. The C1085 gears also show a large amount of cold working and brinnelling. In addition, a small amount of cold material transfer between this combination is evident which should, however, be expected after plastic flow is initiated on the C1085 gears.

Martin hard coated aluminum vs nitrided nitralloy. - These materials successfully complete the tests in vacuum, but fail very quickly in the ambient laboratory environment. The anomaly must result from the environmentally developed physical structure and any difference in physical structure of the Martin hard coated case. Figure 35 shows the gear used in the 21 hr ambient laboratory test, and Figure 36 shows a similar section of a gear tested

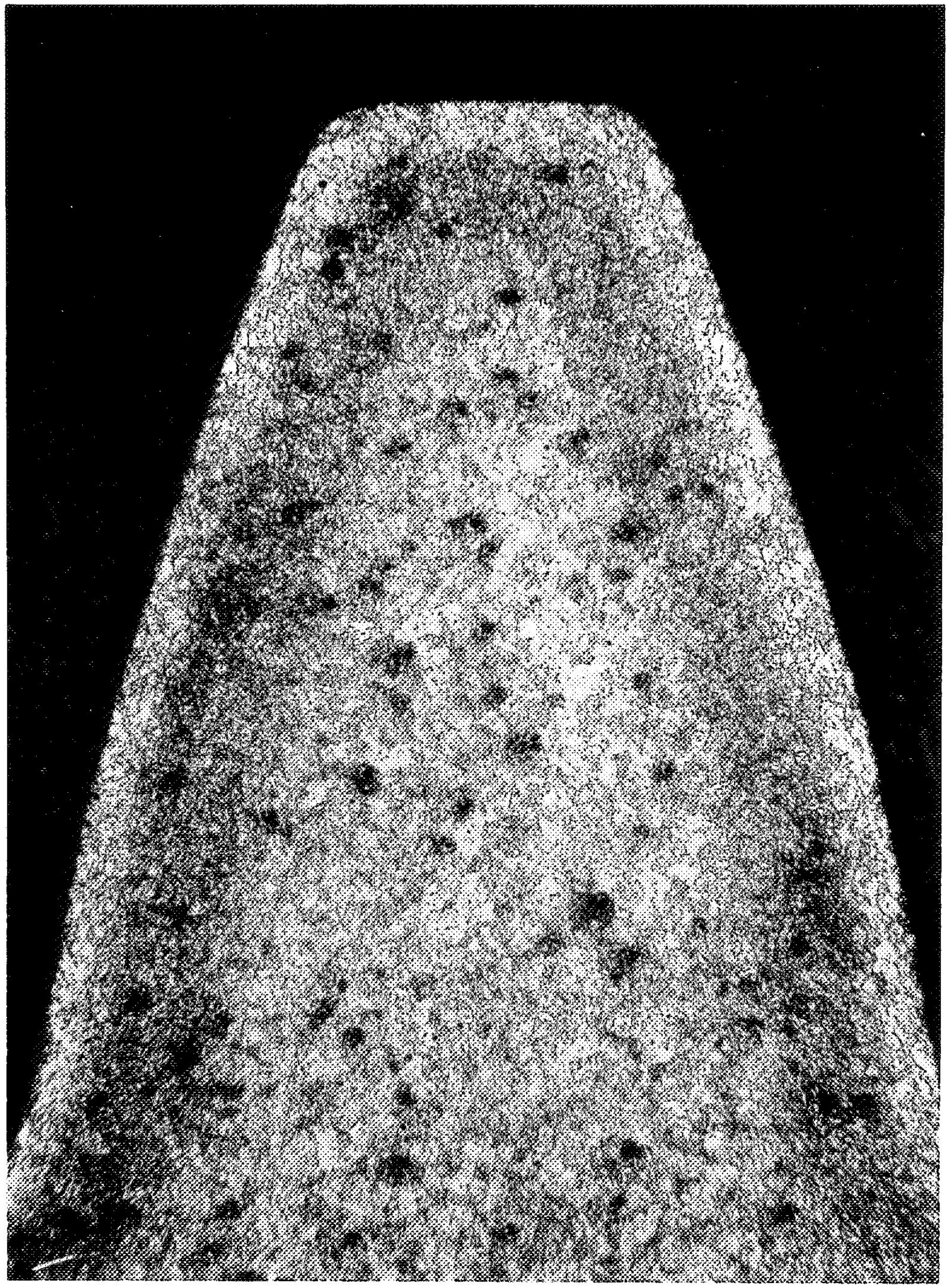
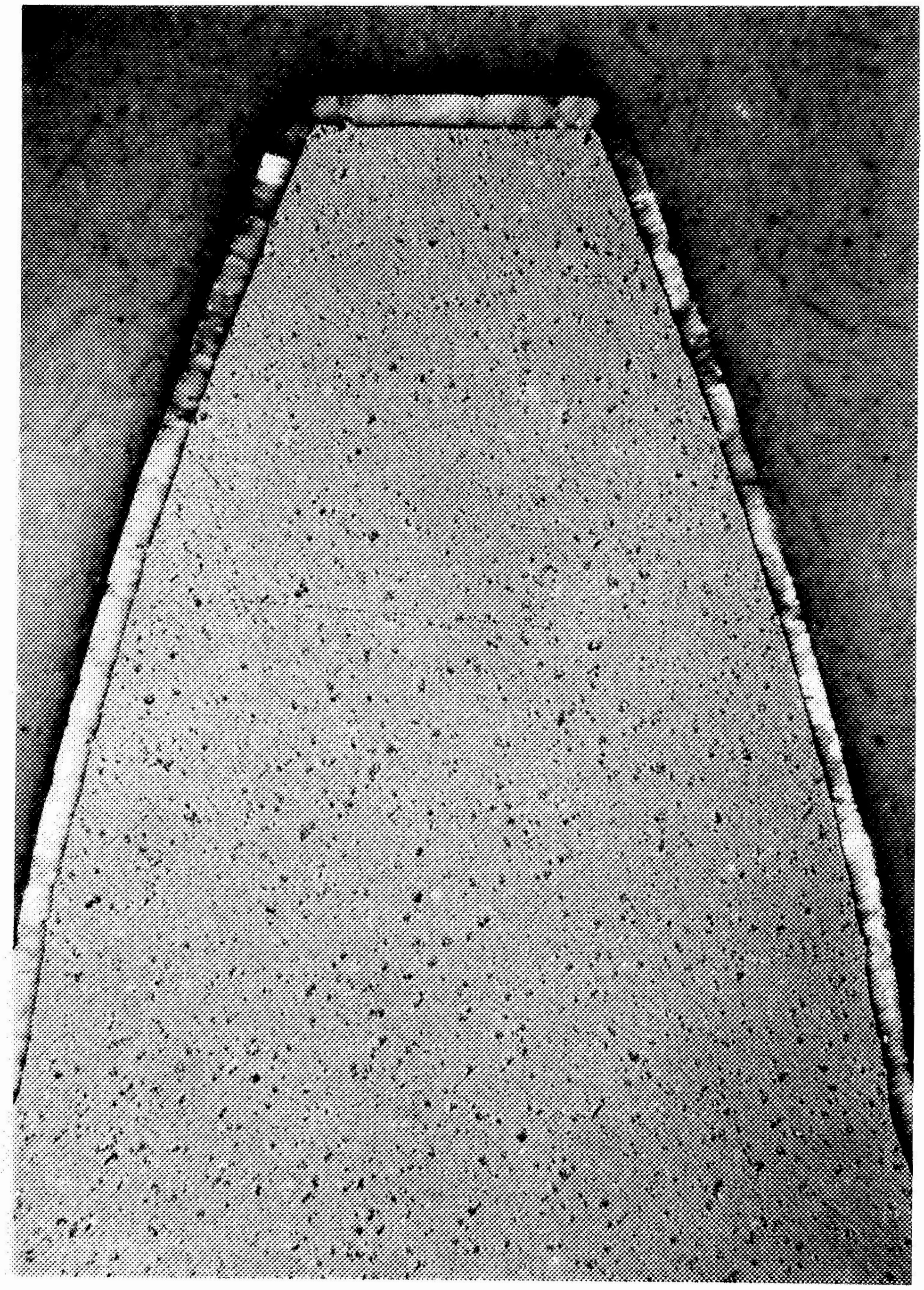


Figure 35 Test 119, Nitrided Nitralloy Involute Gears, MoS_2 Lubricated



100X Magnification

Figure 36 Gear Tooth from Martin Hard Coated Aluminum Involute Gear, MoS_2 Lubricated, Test 119 for 720 hr in vacuum. Each figure shows cracks in the gear hard coat. In the ambient laboratory testing (Figure 35) the cracks are perpendicular to the hardened case and originate at the surface; while in the vacuum atmosphere testing the cracks are parallel to the hardened case and are subsurface in origin.

The reasons for both failures must be clarified by an investigation and analysis based on the surface and subsurface strains due to the Hertzian stresses. Neither the requisite experimental investigation nor the subsequent auxiliary analysis were within the scope of this program. It seems that second order differences in the case depth and composition are responsible for the difference in the abilities of the two gears to withstand the stresses and strains to which they were subjected. It must be apparent that a fully developed technical clarification is necessary to be able to satisfactorily design hardened cases.

One should also consider the original condition of the surface. The anodized surface is composed of a tenacious aluminum oxide ($A\ell_2O_3$) film directly in contact with the aluminum; on top of this film there is a porous surface comprised of $A\ell_2O_3$ and hydrates of the same (refs. 3 and 4). In vacuum the water is removed and only the very anhydrous $A\ell_2O_3$ is exposed to wear. In the laboratory ambient testing the surface is exposed to water vapor in the air. Any hydrates of aluminum oxide can be affected by the water vapor and the frictionally generated temperature increase. Any hydrates developed would wear faster than the anhydrous $A\ell_2O_3$.

<u>Martin hard coated aluminum vs C1085 silver plated and MoS_2</u>. - This combination fails very quickly in atmospheric tests for the same reasons as the previous combination. In vacuum tests the materials perform extremely well. As discussed, the Martin hard coated gears run much better in vacuum and in addition since this coating is porous it allows the silver and MoS_2 to flow into the matrix of the coating. This phenomenon is very evident when the

Martin hard coated gear surfaces are examined with a microscope. In addition, it is interesting to note that very little evidence of plastic flow is found on the C1085 gears in the tests with Martin hard coated material. This is attributable to the reduced frictional energy generated at the mating surfaces.

Nitrided nitralloy vs Phosphor bronze with 15 percent MoS_2 matrix. - Excellent wear characteristics could probably be obtained with this material combination if the strength of the MoS_2 impregnated Phosphor bronze gears could be improved. The combination runs very well in the atmospheric test where the temperature of the gears is slightly reduced. However, when tested in vacuum where the gear temperature is elevated the teeth of the Phosphor bronze gears fail. At the end of 720 hr in the atmospheric tests the Phosphor bronze was just beginning to show evidence of surface fatigue. In all tests with this combination the nitrided gears show MoS_2 smeared or impregnated into the nitrided matrix.

The Phosphor bronze gears must be considered marginal for use at 20 oz-in. torque load, since this load generated a bending stress of ~2000 psi at the root. Test specimens of this material have a rupture modulus of only 7000 to 10 000 psi. Therefore, any use of these gears for long periods of time is subject to fatigue failure. In addition, the contact stress of 60 000 psi would cause surface fatigue unless the strength of the gears could be improved.

Phosphor bronze with 15 percent MoS_2 vs 440C stainless steel.- The results of this combination of materials need no further discussion, since they are almost identical to those obtained with nitrided nitralloy vs Phosphor bronze. The only significant difference is that the Phosphor bronze gears fail by tooth breakage even in the atmospheric tests with this combination.

Martin hard coated aluminum vs Phosphor bronze with 15 percent MoS_2 matrix. - As in all other tests with Phosphor bronze with the MoS_2 matrix, these gears fail by tooth breakage. However, one interesting note with this combination of materials, the layer on the Martin hard coated gears $A\ell_2O_3$ does not fail in the atmospheric tests after 307 hr of usage. This suggests that the $A\ell_2O_3$ coating accepts the MoS_2 lubricant into the coating and as a result the surface temperature fluctuations of the mating gears are reduced.

Light anodized aluminum vs beryllium copper. - This combination fails in both atmospheric and vacuum tests due to the breakdown of the very thin coating of $A\ell_2O_3$ which is incapable of supporting the contact stresses. Another interesting occurrence is the marked transfer of beryllium copper to the light anodized aluminum gear in vacuum. However, this should probably be anticipated because of the apparently high surface temperatures encountered during these tests at 20 oz-in. of torque load.

Martin hard coated aluminum vs beryllium copper. - These materials fail in the same manner as the previous combination. Again a marked transfer of material occurs in the vacuum tests. This transfer is initiated in vacuum before the $A\ell_2O_3$ coating fails.

Light anodized aluminum vs Martin hard coated aluminum. - This combination fails in a very short period of time due to the failure of the light anodized coating. The failure is, of course, more rapid in the atmospheric tests.

Martin hard coated aluminum vs Phosphor bronze with 15 percent MoS_2 matrix. - Breakage of the teeth on the Phosphor bronze gears is the cause of failure for this combination.

Nitrided nitralloy vs beryllium copper. - This combination runs very poorly. The vacuum tests are terminated when the torque increases beyond the capability of the dc motors to turn the rig. The atmospheric tests are terminated when the teeth fail on the beryllium copper gears after a high percentage of wear occurs.

Examination of both vacuum and atmospheric test samples shows marked adhesion and material transfer in all tests.

Second Test Series

The second series employed expanded torque loads (10, 20 and 30 oz-in.) and speeds (900 and 1800 rpm). The summarized results, presented in Table VII, indicate that the nitride enrichment of these gears (Figure 12) causes brittleness. Only 4 of the 20 tests conducted during this series ran 720 hr without chipping of the nitrided case. In each of these four tests the torque load was only 10 oz-in. Because of the problem of the brittle case and the subsequent chipping of the nitrided gears, no correlation between gear life, load and/or speed can be made.

Cycloidal Gear Tests

The results obtained from the cycloidal gear tests (Table X) only confirm the known disadvantages of this system of gearing. The change in center distance which results in the loss of conjugate action with the cycloidal gearing system is caused by thermal expansion as indicated by the additional backlash required to operate. This loss of conjugate action combined with the less precise (unground) gear profile causes increases in both contact and shear stresses.

CONCLUSIONS AND RECOMMENDATIONS

In the unlubricated tests, one combination of gear materials, nitrided nitralloy and 440C stainless steel, perform well in both vacuum and atmosphere environments with contact stresses as high as 50 000 psi. This use capacity is of considerable importance when one considers the necessary prelaunch testing. Nitrided nitralloy and carburized C1020 showed promise, but it was erratic.

The results of the lubricated gear tests indicate that with contact pressures of 40 000 psi at the lubricated surfaces, one cannot safely expect a dynamic life of over 200 hr. These gears last for over 200 hr with very little wear and then fail in a period of less than 24 hr. Obviously, it is impossible to predict gear life on the basis of wear at one set of conditions.

Of the unlubricated gears, only those which last 720 hr are considered; with these the wear is linear with time. The wear rate, of course, is greater at the higher torque loads.

Clearly, further studies should be conducted. As in virtually all wear experiments there is considerable scatter in the relatively good abrasive wear resistance in both vacuum and air when the case depth corresponds to that recommended (ref. 18) and, accordingly poor wear characteristics when the depth exceeds the recommendation. Chipping occurs in the thick case. It is therefore recommended that further work be devoted to the study of nitrided case thickness effects, possibly with a matrix other than nitralloy.

Further work should be devoted to the full explanation of the different behavior of Martin hard coated aluminum in vacuum and air, particularly when aluminum electrical, thermal and weight properties are considered. Also, the mechanical failure effects of both Martin hard coated aluminum and nitrided nitralloy should be investigated. The Hertzian stresses and strains exhibited within the hard case and at the junction of the case and the matrix should be explained. Ideally, one should be able to predict at least in general terms how and why a material combination would fail in the mechanical sense of failure.

The rather surprising result that three types of solid film lubricant, all MoS_2 based, should catastrophically fail in less than 24 hr after running for more than 200 hr requires further investigation. In addition, the Phosphor bronze MoS_2 filled gears should be investigated further. The gear should be analytically designed for its load condition, which could require tooth widening.

APPENDIX A

VACUUM GEAR TEST APPARATUS ASSEMBLY PROCEDURE

Following is a detailed assembly procedure for all components except the spline shaft assembly in the master gear index drum, and the transducer and bearing assembly in the S-shaped master gear indexing bracket.

BASE PLATE AND MASTER GEAR MECHANISM ASSEMBLY

Assemble all bearings (top and bottom) on the base plate. Use special thin retainer rings on the bottom bearings. (The top bearings need not have retainer rings.) The bearing shields on the top bearings must be on the top side of the bearing block; and on the bottom bearings, must be on the bottom side of the block.

Assemble the master gear indexing drum with spline shaft in the center of the base plate. Note: two of the four bolts used in assembly are 10-24 with collars around them as locating pins. The other two are 10-32.

Slide S-shaped master gear index bracket over the spline shaft with the set screw in the bracket facing the flat on the shaft (Figure 37). Tighten the top nut over the shaft until all play in the S-shaped bracket is removed. Rotate the bracket to make sure it indexes without binding on index drum. Tighten the set screw on the spline shaft.

Assemble the bearings and retainer ring in the U-shaped master gear brackets as shown in Figure 38 with bearing shields on the outside. Assemble master gear shaft, master gear, spacers and lock nuts as shown in Figure 38 and load bearings lightly to remove play using spring washer.

Mount both U-shaped brackets (spring-loaded) on S-shaped bracket as shown in Figure 39. Make sure the bracket travels freely on the ball bushings without binding. During assembly, make sure both master gears are so mounted that the gear clamp is on the top side of the gear. Adjust and fasten master gears on the shaft so that the top face of the top master gear is 4.569 in. from the top surface of the base plate; and the top face of the bottom master gear is 3.264 in. from the top surface of the base plate.

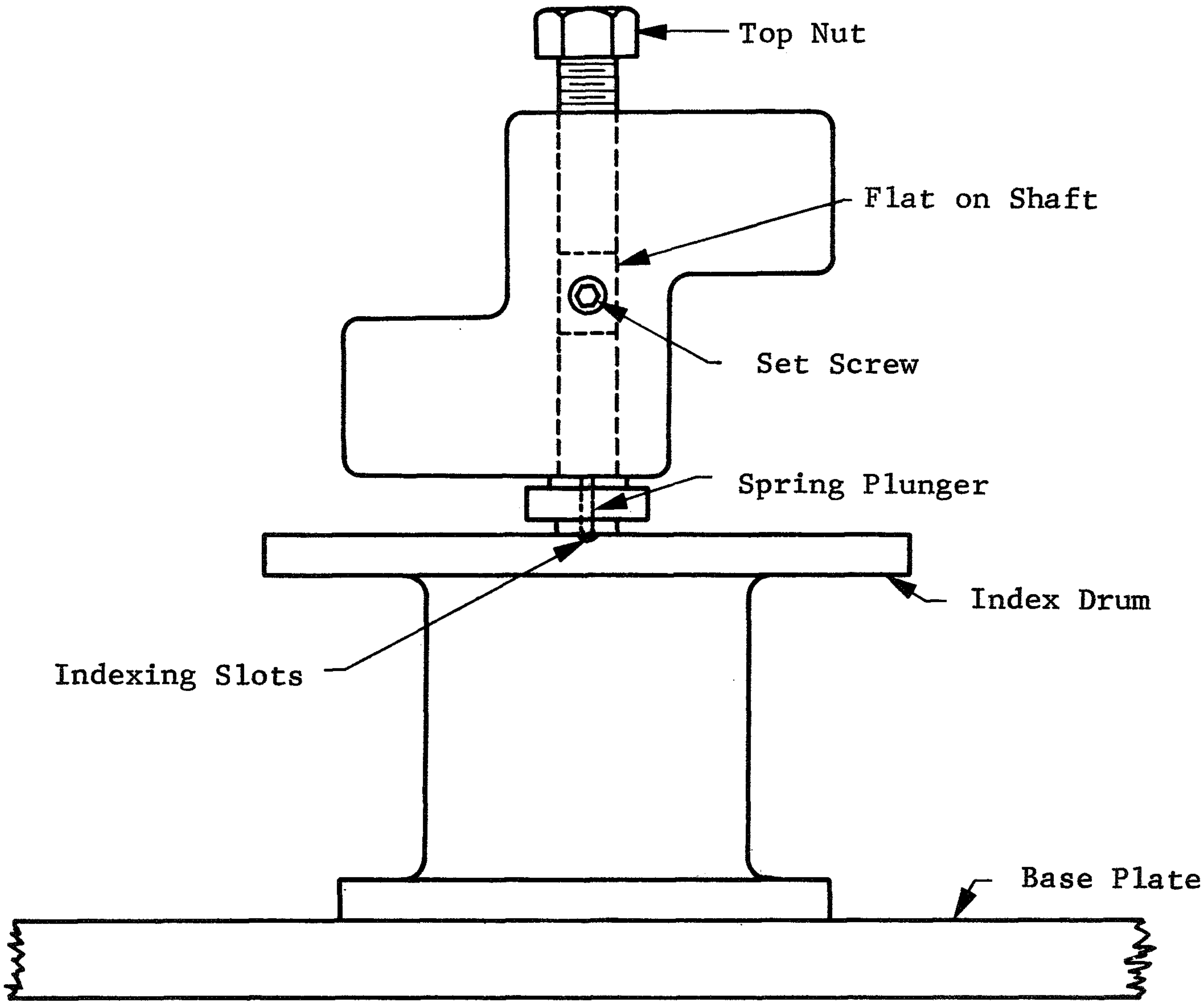


Figure 37 Master Gear Indexing Bracket

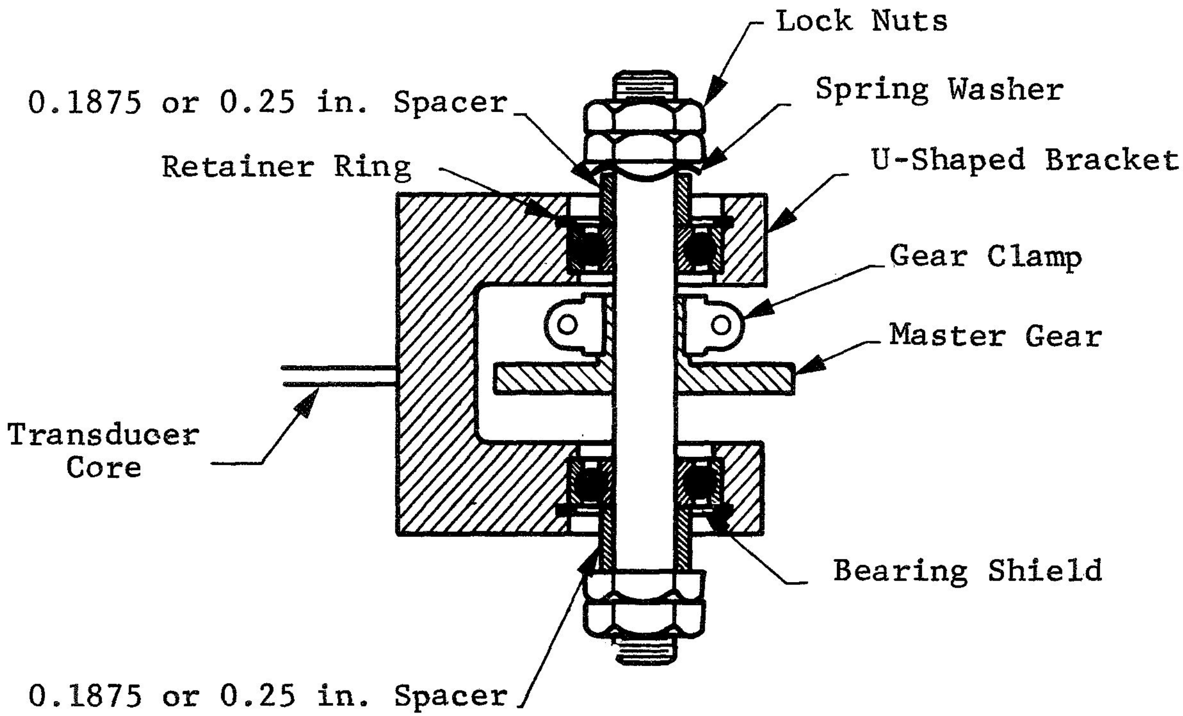


Figure 38 Bearing and Retainer Ring Assembly

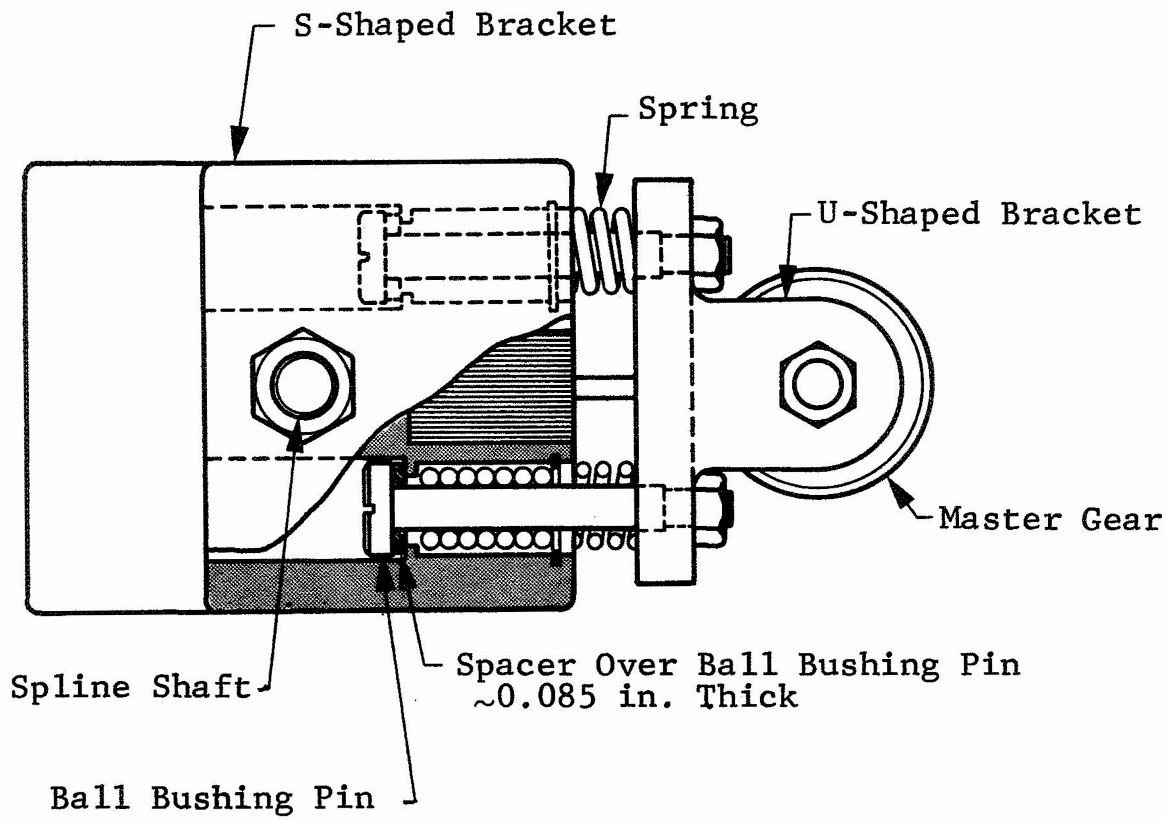


Figure 39 Master Gear Mounting Assembly

ASSEMBLY OF FOUR-SQUARE TEST RIGS

Press fit all six bearings in the test rig as shown in Figure 40 with retainer rings. Note: observe shield locations as per Figure 40. The parts in Figure 41 are identified below.

1. Drive shaft coupling
2. Collar 0.60 in. long
3. Bearing shield disk
4. 0.1875 in. gear, Material I
5. Gear clamp
6. Gear clamp
7. Drive shaft
8. 0.1875 in. gear, Material II
9. Bearing block, upper right
10. Bearing block, lower right
11. Collar 0.1875 or 0.25 in. long
12. Spring washer
13. Bushing
14. Bushing
15. Upper left shaft
16. Spring washer
17. Collar 0.1875 or 0.25 in. long
18. Upper left bearing block
19. Split bushing
20. Split bushing clamp (same as gear clamp)
21. Gear clamp
22. 0.125 in. gear, Material I
23. Torque load coupling
24. 0.125 in. gear, Material II
25. Gear clamp
26. Split bushing clamp (same as gear clamp)
27. Split bushing
28. Lower left bearing block
29. Collar 0.875 or 0.25 in. long
30. Spring washer
31. Shaft, lower left

Assembly of Parts 1 through 13 of Figure 41

Push Shaft 7 through bearing blocks 9 and 10 placing gears 8 and 4, gear clamps 5 and 6, and disk 3 in position.

Locate shaft so that it extends about 0.5625 in. outside upper face of bearing block 9.

Plate collar 2 and clamp coupling 1 should be in position as shown.

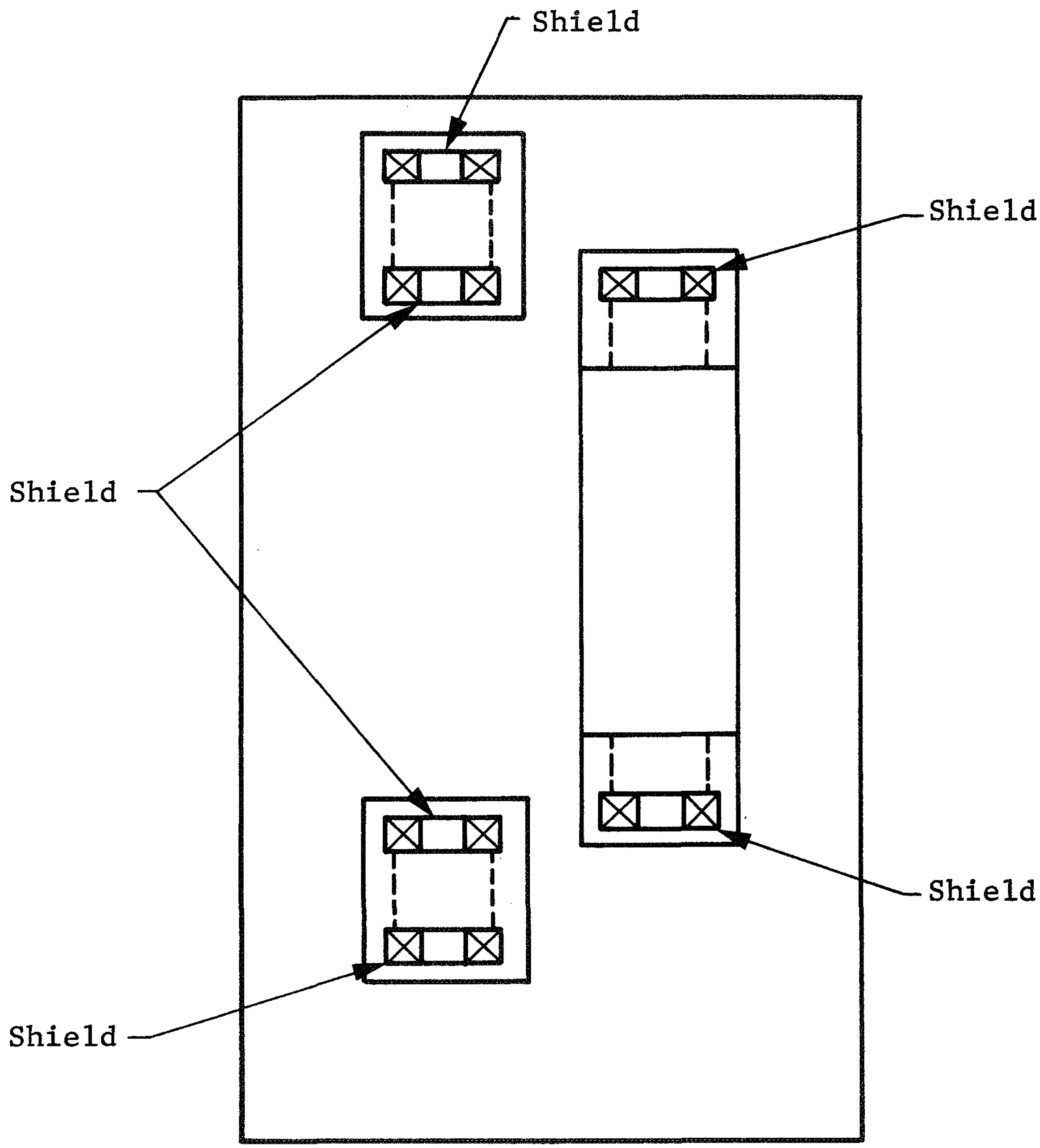


Figure 40 Front View of Test Rig with Bearing Locations

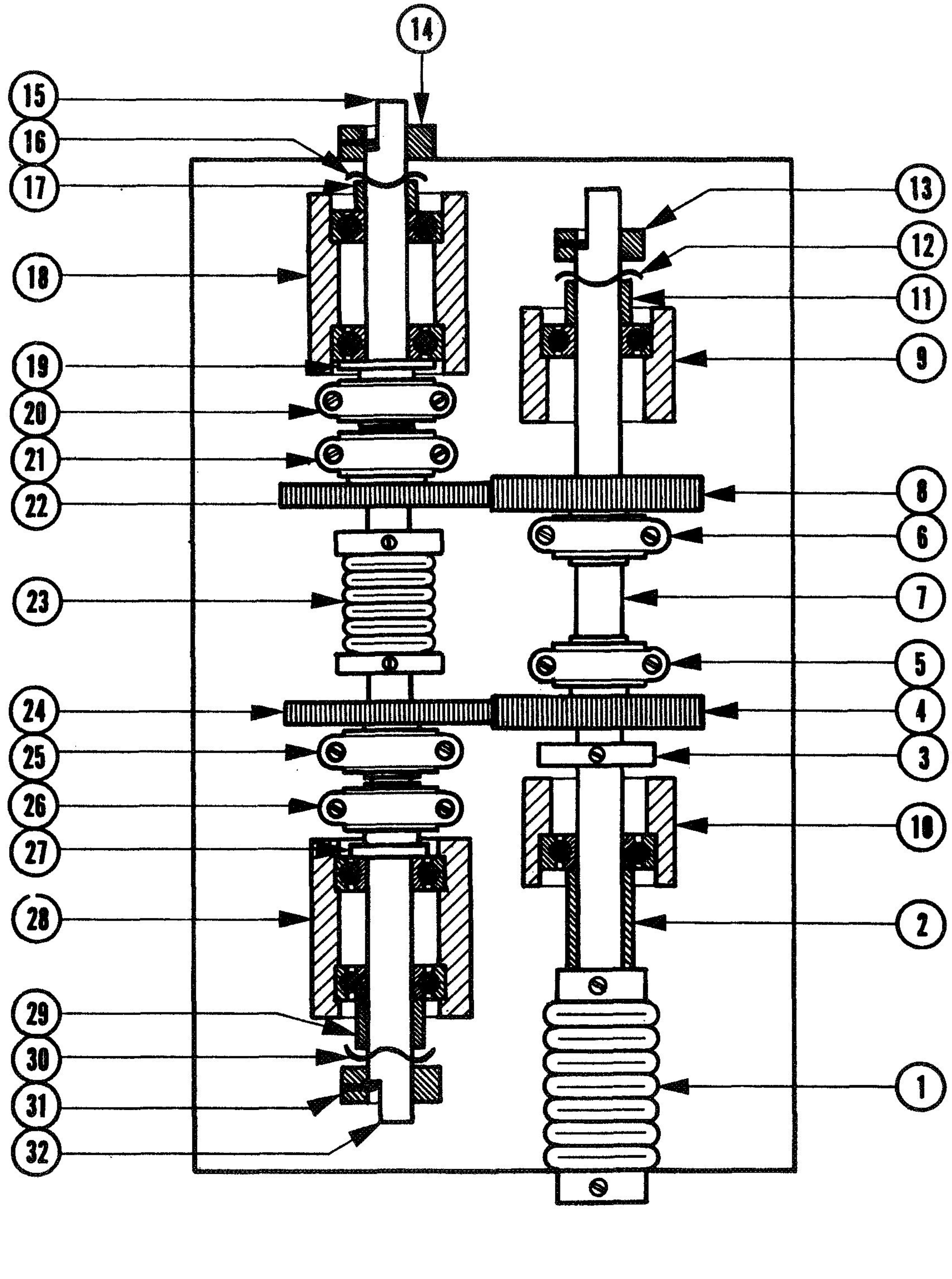


Figure 41 Front View, Test Rig Assembly

Slip on collar 11 at upper end.

Slip on spring washer 12.

Clamp on bushing 13 after compressing spring washer 12 (one-half total deflection) thereby loading both bearings.

Clamp disk 3 on shaft so that clearance between lower face of disk 3 and upper face of lower right bearing block 10 is only a few thousandths of an inch.

Temporarily clamp lightly gear clamps 6 and 5. Exact location of gears on shaft 7 will be discussed later.

Assembly of Parts 14 through 32 of Figure 41

Push shaft 15 through both bearings in bearing block 18.

Slip on split bushing 19 and split bushing clamp 20 on shaft 15.

Locate shaft 15 so that it extends 0.5625 in. outside upper face of bearing block 18.

Clamp on split bushing clamp 20 adjacent to bearing.

Slip on collar 17.

Slip on spring washer 16.

Slip on bushing 14.

Apply axial load on bearings by compressing spring washer 16 (half-full deflection) and lock bushing 14 in place.

Slip on gear 22 and gear clamp 21. Slip on torque load coupling 23 and attach it temporarily to lower end of shaft 15.

Locate gear 24, gear clamp 25, split bushing 27 and split bushing clamp 26 as shown in Figure 41 and slip shaft 32 through bearings in bearings block 28 and upward all the way through to coupling 23.

Position shaft 32 so that 0.5625 in. extends outside the lower face of bearing block 28.

With shaft 32 in this position and split bushing 27 touching upper bearings, clamp on split bushing clamp 26 on shaft.

Slip on collar 29.

Slip on spring washer 30.

Slip on bushing 31 and load bearings between split bushing 27 and bushing 31 by deflecting spring washer to half-full deflection and clamp bushing 31 in place.

Mount coupling 23 on shaft 32 leaving 0.9375 in. axial movement for gear 24.

Mount coupling 23 on shaft 15 leaving 0.9375 in. axial movement for gear 22.

ASSEMBLY OF FOUR-SQUARE TEST RIGS ON BASE PLATE

Place two locating pins in base plate for test rig location.

Place 0.285 in. collar on top of top bearing in base plate (see Figure 42).

Position and clamp test rig on base plate with two 10-32 bolts. Use both locating pins during mounting and remove all loose pins after mounting so that they do not fall out due to vibrations during test.

Slip magnet drive shaft through the bottom of the base plate and attach drive shaft coupling 1 to top end of magnet drive shaft.

Slip on 0.080 in. collar on the bottom end of shaft.

Slip on magnet with clamp.

Push up magnet and clamp magnet on the shaft at the same time pushing coupling 1 toward base plate so that no axial play in magnet drive shaft results after assembly. Note: do not load bearings.

POSITIONING GEARS

Locate and clamp gear 22 (Figure 41) so that the top face is 4.574 $\pm$0.002 in. above the top face of the base plate.

Locate and clamp gear 24 (Figure 41) so that the top face is 3.269 $\pm$0.002 in. above the top face of the base plate.

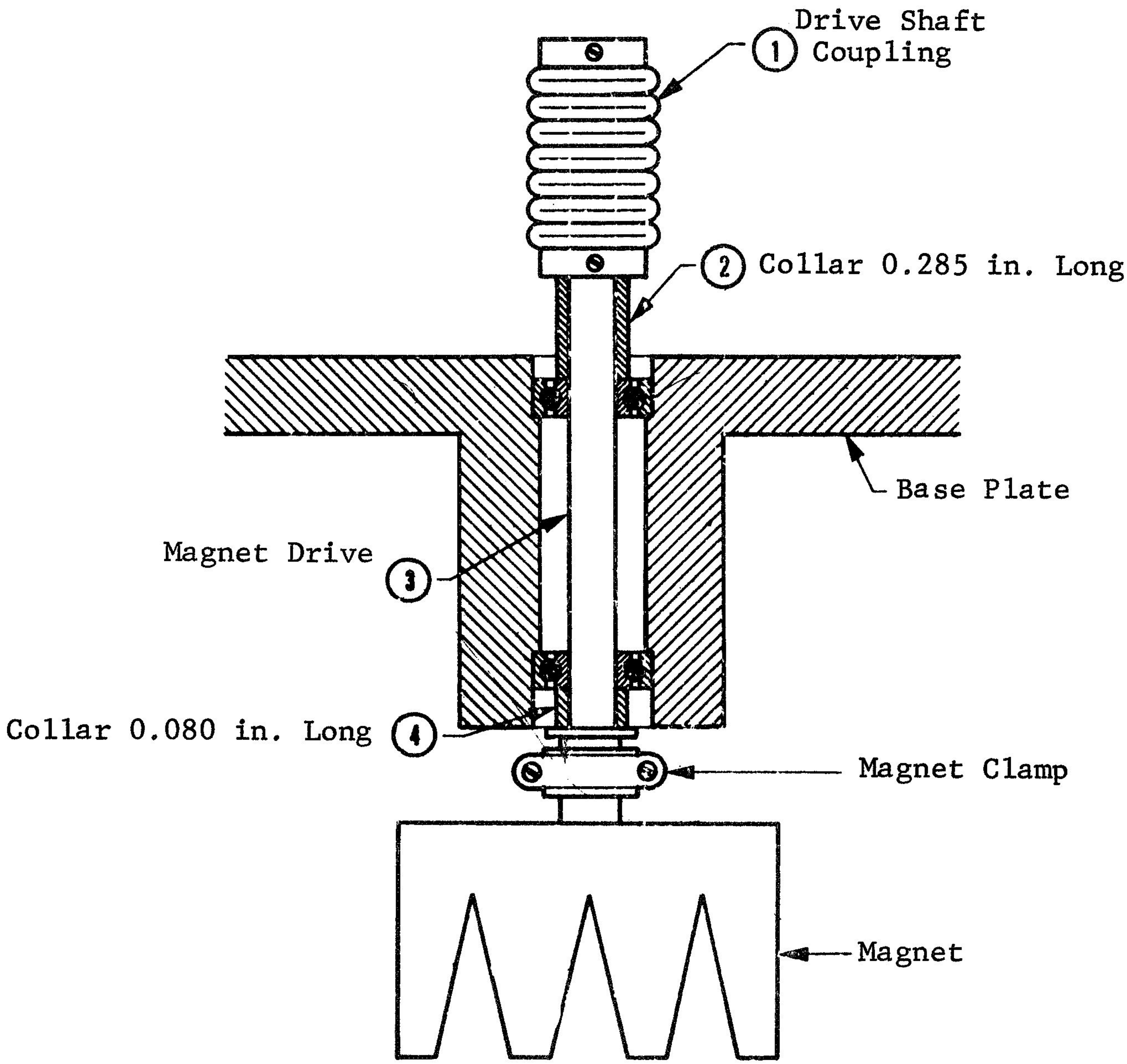


Figure 42 Magnetic Drive Assembly

Visually locate and clamp gear 4 (Figure 41) so that the centerline of gears 24 and 4 coincide.

Visually locate gear 8 (Figure 41) so that the centerline of gears 22 and 8 coincide.

Clamp gear 8 lightly on the shaft so that it can turn freely, but not slide out of its axial location.

Mate master gears with all test gears and verify location accuracy.

LOADING THE TEST RIGS (FIGURE 41)

Attach torque wrench to upper end of shaft 15 (Figure 41).

Locak shaft 32 (Figure 41) by placing Allen head wrench in bushing 31 and using the same as a stop arm against test rig frame, while shaft 15 is turned by the torque wrench.

Apply torque load as specified on shaft 15. During load application gears 24 and 4 will remain stationary and gear 22 will rotate with shaft 15 while gear 8 will rotate on stationary shaft 7.

When full load is applied, clamp gear clamp 6 tightly on shaft 7. Visually check alignment of gear 8 with gear 22.

CHECKING THE TORQUE SETTING

Lock shaft 32 as discussed earlier. Gradually apply load with torque wrench on shaft 15. All gears will remain stationary during this load check procedure. However, by careful observation of gear 22 (Figure 41) very small movement of gear 22 (movement equal to backlash) will be noticed in the vicinity of actual torque. This torque value must be written ± 2 oz-in. of the specified torque. If the torque is not within tolerance, it must be reset. There will be no movement of gear 22 above and below the actual load value.

FINAL CHECK ON ASSEMBLY

Check all gears and make sure that they are on correct location and mating against the desired gears. Record gear locations and test rig numbers on data sheet. Rotate master gear mechanism and mate master gears against every test gear and rotate the respective test rigs to make sure they turn without excessive binding.

Go over all clamps and make sure all screws are absolutely tight. Take vacuum cleaner and remove all loose lint (from gloves) from the test apparatus.

Prepare the vacuum system for test; in other words, have the system cleaned out and new gasket made and VacSorb pumps baked out.

Lower the apparatus into the system very slowly so as not to damage spline coupling. Use the fixture for this operation. Locate the test rigs to match with their respective motors.

Set the master gears in their neutral position. Locate all signal leads and make sure you can identify them as to their soldering location on feedthrough.

Replace top flange and mount feedthrough after carefully soldering leads to it. Make sure the soldering iron is clean and do not use any flux; use special fluxless solder.

Before tightening up large flange, check the output from master gear transducers and take a few readings on gears.

Tighten up the top flange, check the output from master gear transducers and take a few readings on gears.

Tighten up the top flange and pump down the system.

In Figure 42, Item 3 is a bellows type coupling. The bellows is a spring which transmits a dynamic load to the shaft. It should be replaced with a coupling that minimizes dynamic load transmission.

APPENDIX B
WEAR MEASUREMENT FOR VACUUM TESTS

The wear measurement technique selected is basically similar to the approach used by most gear manufacturers for determination of the precision of new gears using master gears. This technique is described in many gear textbooks and manufacturer's catalogs and is better known as total composite error (TCE) chart recording. The total composite error represents the peak-to-peak variation in the center distance between a test gear and a reference master gear. With the use of a special master gear, we also monitor wear on the test gears by the measurement of changes in the center distance.

On all gear tests, the criterion for termination is based on either a total elapsed time of 720 hr or ~10 percent wear at the pitch radius. The gears used have a tooth thickness of 0.0327 in. at pitch radius, 10 percent wear refers to reduction of this dimension by 0.0032 in., or a wear of 0.0032 in. at the pitch radius. In other words, 1 percent wear represents a wear of only 0.0003 in. at the pitch radius.

Assuming as a first approximation that as a gear wears the pressure angle at the pitch line does not deviate much from the original 20 deg, 10 percent wear or 0.0032 in. reduction in tooth thickness would be reflected as a center distance of 0.0046 in. From past data it is known that termination at a change of 0.0055 in. results in better correlation.

The solid-state displacement transducers in the master gear arrangement provide millivolt signals linearly proportional to displacement, and the termination criterion used in the output in millivolts rather than displacement in inches. As will be explained later, the top transducer gives an output of 120 mV for a change of 0.0055 in., and the bottom transducer gives an output of 110 mV for the same change in center distance. In other words, an 11 to 12 mV change in output represents ~1 percent wear. Transducer calibration should be repeated when the vacuum system is opened.

The repeatability of test data has been established as ± 20 mV on overall gear wear readings, which means that at any time the gear wear prediction could be in error by $\sim \pm 2$ percent. This percentage is acceptable as termination criterion but inadequate for accurate prediction of wear rates, especially during the early stage of the test.

INSTRUMENTATION

The vacuum test apparatus is so designed that all test gears are approximately the same distance from the vertical centerline axis of the apparatus. Furthermore, half of the test gears are assembled so that they are within ± 0.002 in. in one horizontal plane (top); the other half being similarly positioned in a second horizontal plane (bottom). For each plane, there is a spring-loaded master gear arrangement again precisely assembled so that during wear measurements the master gear mates with each test gear within the wear track.

Two solid-state transducers (Sanborn Type 7DCT-050, LVDT Model B12-823-2P11) are mounted in the master gear mechanism. The top transducer records the displacement of the master gear in the top plane and the bottom transducer correspondingly records displacement in the bottom plane. These transducers are calibrated for a 6 V dc input. The two input leads are color coded (red and black). Each transducer has two output leads (green and yellow) and the output signal (absolute value) is dependent on the location of the core within the transducer. This absolute value of the signal must not be confused with the millivolt reading representing wear on the gear. The measured change in millivolts of this output signal represents the gear wear.

The input and output leads from both transducers are brought to the eight pin vacuum feedthrough in the top flange as shown in Figure 43 and are externally connected through a control panel to the recorder. The panel contains a battery power source, voltage

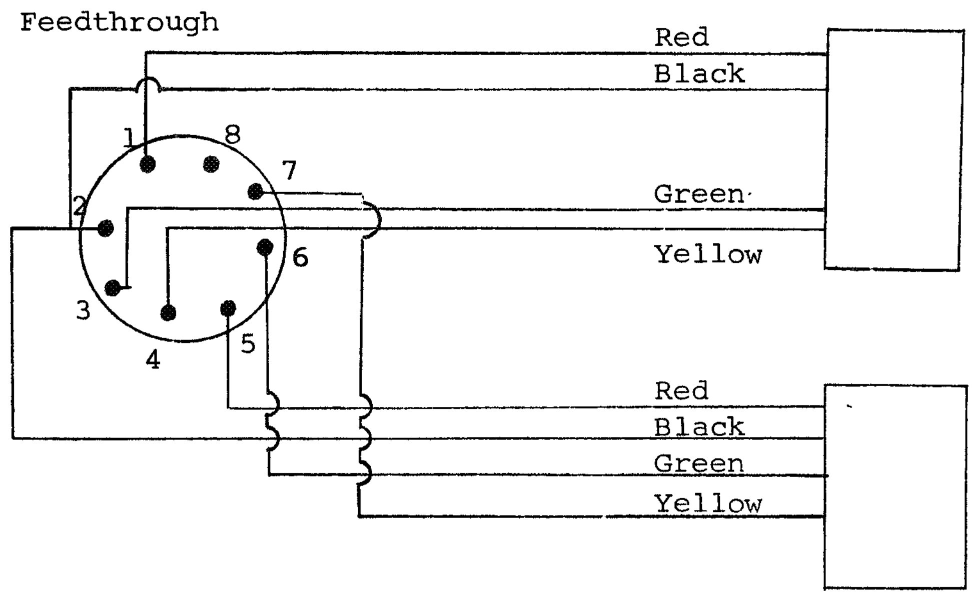


Figure 43 Transducer Schematic

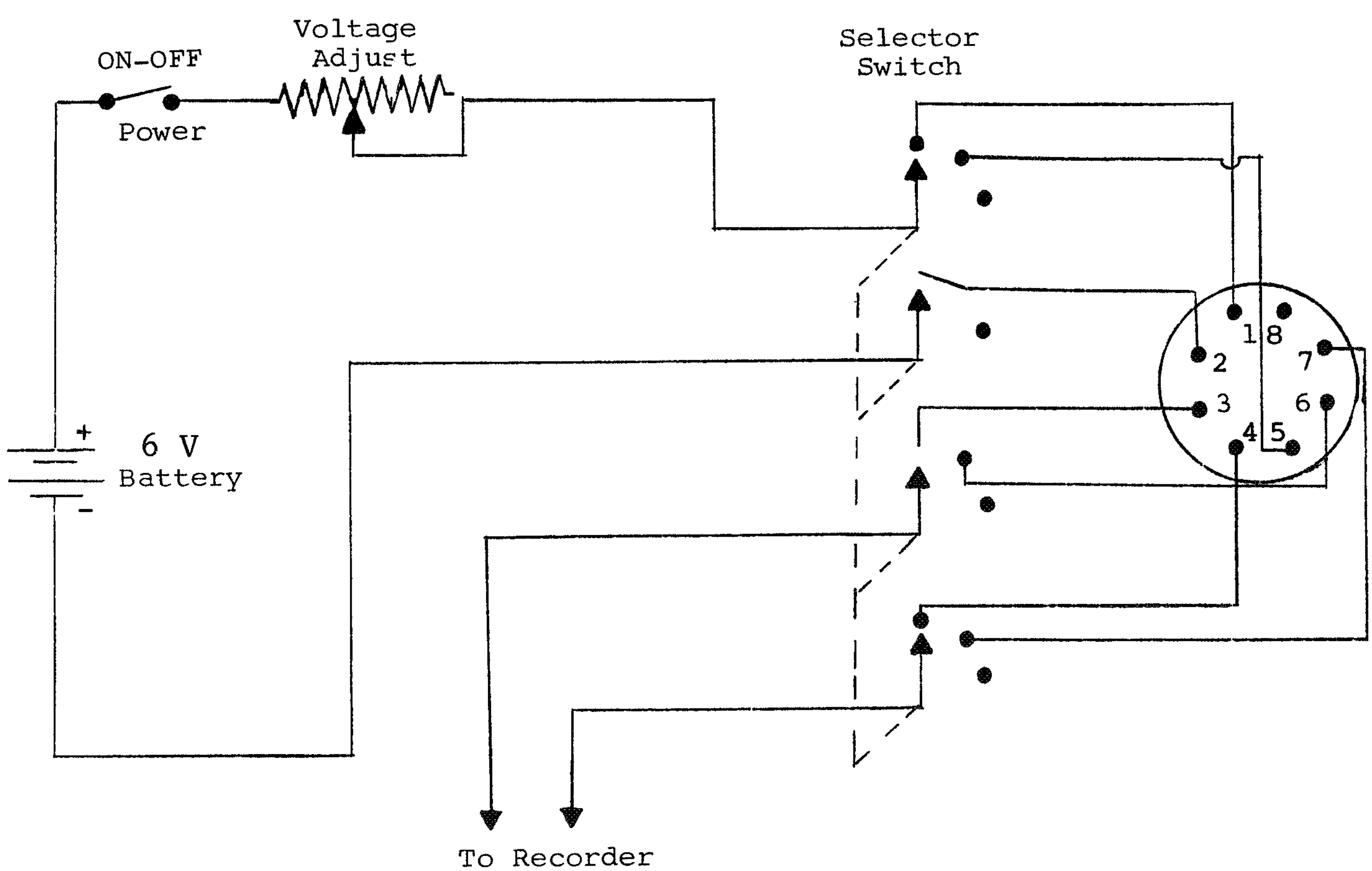


Figure 44 Control Panel and Recorder Schematic

adjustment selector switch for connecting the desired transducer to the recorder, and a power on/off and shorting switch. The circuit for the control panel is shown in Figure 44.

METHOD OF MEASUREMENT

Before a procedure for data testing can be outlined, it is necessary to understand the working of the master gear arrangement. The master gears have a set of teeth comprised of two or three adjacent teeth at one location of the circumference of a rectangular shape having a tooth thickness of 0.0375 in. About 180 deg from this location there is another set of teeth, completely filled in so that during wear measurement at this location the master gear will ride on top of the teeth of test gears instead of meshing. The technical reason for the arrangement has been fully discussed in the main body of the report.

One measurement is taken of the center distance when the desired square tooth engages with the test gear; another reference measurement is made when the filled in teeth described ride on top of or on the outer diameter of the test gear. The difference between these two measurements is the actual overall measurement for the particular test gear.

By recording this measurement for each gear periodically and noting the changes therein, the amount of total wear on the gear can be predicted by the following relationship:

$$\text{percent wear} = \frac{\text{change in overall reading(millivolt)}}{\text{calibration (millivolt/percent wear) of transducer coefficient}}$$

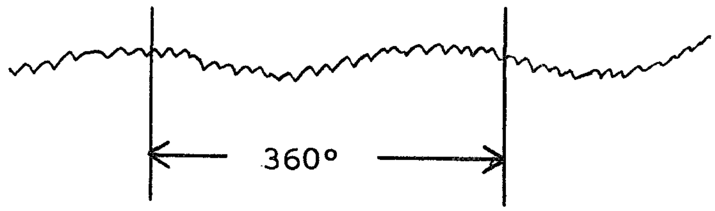
calibration coefficient of top transducer = 12 mV percent wear

calibration coefficient of bottom transducer = 11 mV percent wear

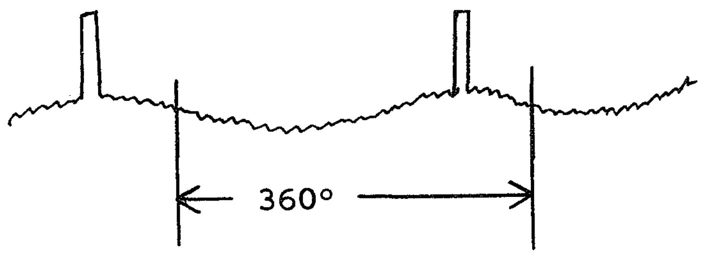
The type of recording one should expect can be best explained by a series of illustrations. Let us first consider a TCE chart where a standard master gear is used to check a new test gear. As the test gear, which is spring-loaded by a master gear, rotates, the variations in center distance recorded on chart paper would look like Figure 45a. The sine wave variation over the 360 deg rotation gives the overall runout, whereas the small hash represents tooth-to-tooth variations.

If the same master gear had one or two adjacent teeth especially made, say of rectangular shape, so that they penetrate a little further in the test gear compared to the remaining teeth, the curve would look like Figure 45b. If in addition to the rectangular teeth, the same master gear had a couple of teeth filled in, the curve would look like Figure 46. The curves produced by the master gear used in this study are quite similar to those of Figure 46 but have more peaks produced by more than one set of rectangular teeth. The top master gear produces the recordings of Figure 47a; the bottom gear corresponds with Figure 47b. The extra peaks are due to undesirable rectangular tooth widths. The points of interest in Figures 47a and b are:

- AB - The millivolt reading where A and B are average hash locations at the start of the curve.
- EF - A millivolt reading similar to AB which in most cases will not differ more than 10 mV from AB. Points E and F are average hash locations at the end of the curve.
- CD - The millivolt reading representing rectangular tooth travel, D being the extreme top of the peak and C the average hash at that location.



(a) Standard Gear



(b) Modified Gear

Figure 45 Representative TCE Charts for Standard and Modified Gear

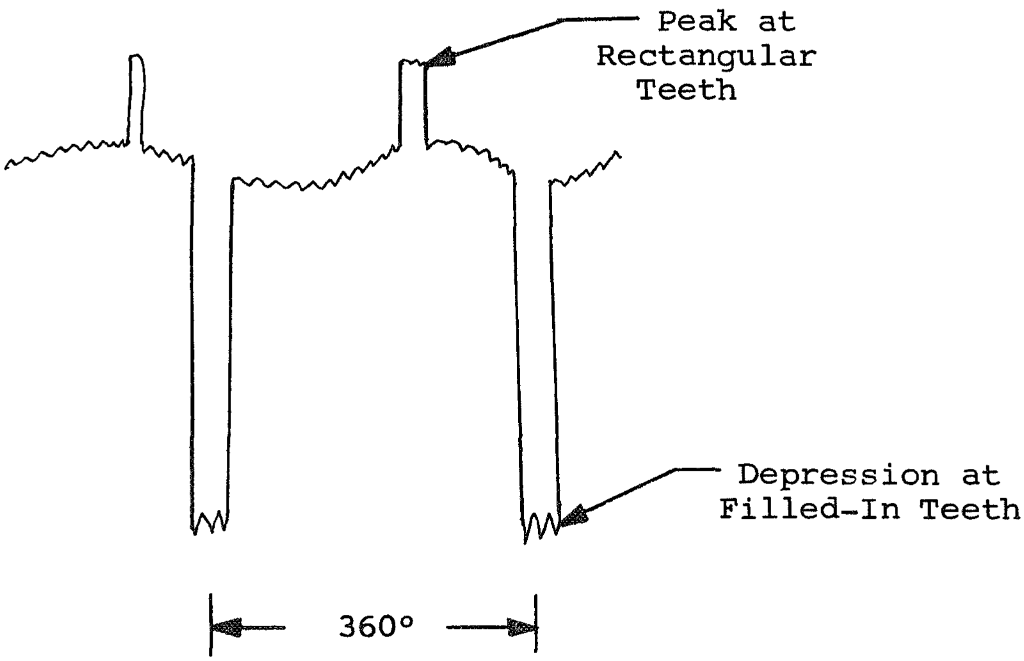
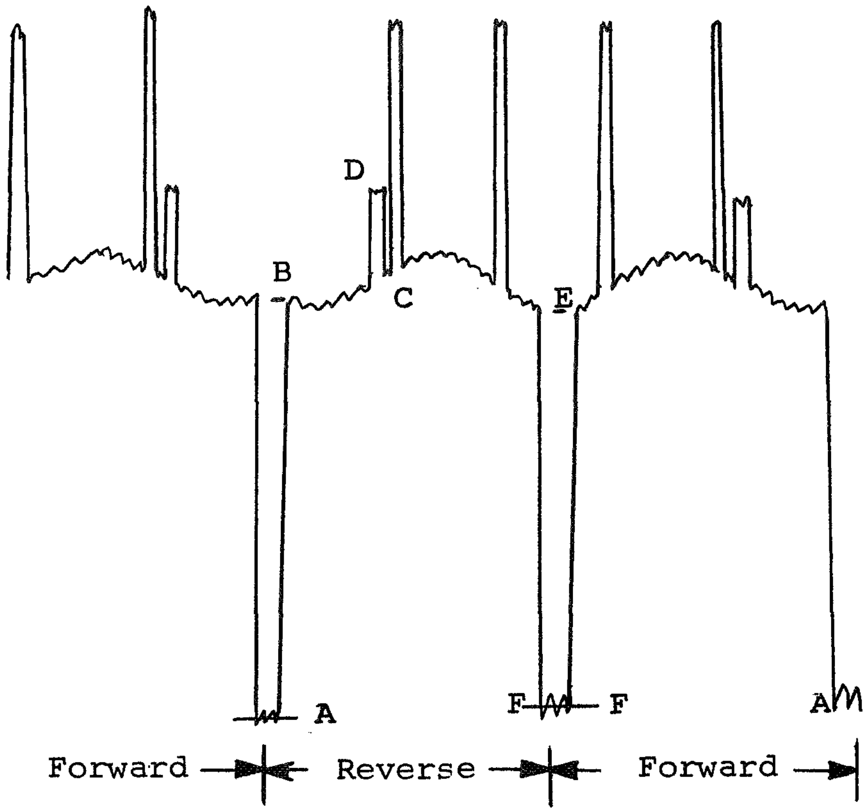
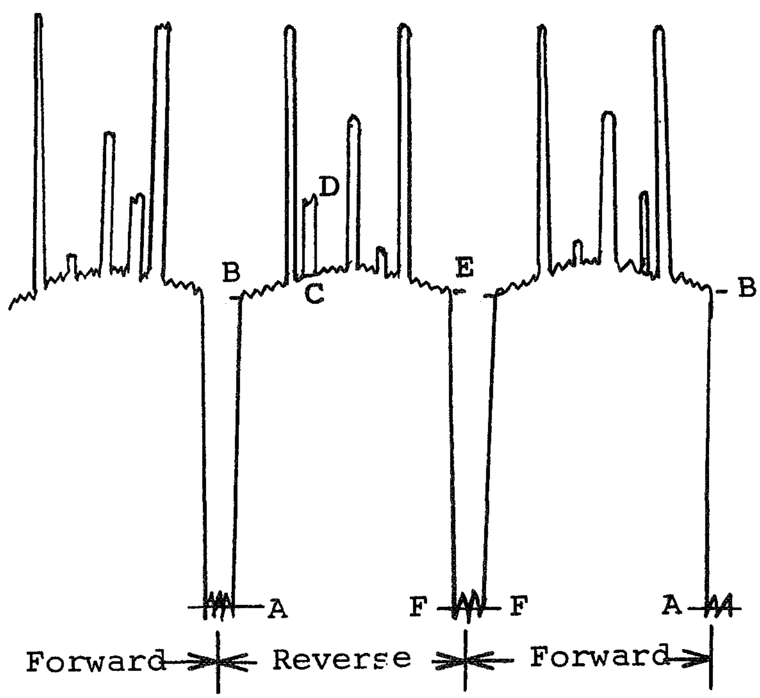


Figure 46 Representative Curve for Modified Master Gear with both Rectangular and Filled-In Teeth



(a) Upper Modified Master Gear



(b) Lower Modified Master Gear

Figure 47 Representative Wear Charts Showing Upper and Lower Modified Master Gear

The overall measurement in millivolts for the particular gear is tabulated as:

$$\left[\frac{AB + EF}{2} + CD\right]$$

Step-By-Step Data Taking Procedure

(1) At the start of test make sure the master gear mechanism is in a neutral position so that the bottom master gear is between Fixtures VII and VIII, and the top gear is between Fixtures IV and III.

(2) Learn to identify the gears on each test fixture. There are four gears on each test rig, two on top and two on bottom. Looking from top center of the test apparatus at any one test rig, two of the four gears will be in clockwise location with respect to the center of that test rig, and the other two will be in counterclockwise direction. For example, on Fixture VII we can identify all four gears as: top CW, top CCW, bottom CW and bottom CCW.

(3) Turn on the power switch on the control panel.

(4) Adjust the supply voltage to 6 V.

(5) Set the ac coupler on the recorder to:

Time Constant - dc
High Frequency - 3
Sensitivity - 100 mV/cm

(6) Set the right channel on the recorder amplifier to:

HI - out
Sensitivity - XI
Paper Speed - 1 mm/sec

(7) Turn master gears clockwise (looking from top) one index position so that top master gear mates with top CCW gear on Fixture III, and bottom master gear mates with bottom CCW gear on Fixture VII.

(8) Turn selector switch on control panel to connect bottom master gear in circuit (Position 3).

(9) Switch recorder to limiting position. This prevents the pen from overshooting at any time and causing damage.

(10) Rotate magnet on Fixture VII and adjust zero setting knob on the right channel of the recorder so that the pen records in the center of the paper.

(11) Try to bring the curve into correspondence with that of Figure 47a by moving the magnet forward and backward.

(12) Once the curve is thus identified, readjust the zero setting so that points B and E fall along the middle of the chart paper (range of right channel).

(13) Switch the recorder to "operate" position.

(14) Slowly continue to turn the magnet in a forward direction until point A (Figure 47a), is reached, reverse the magnet and continue turning slowly through points B, C, D, E, and F. At position F reverse the magnet again and to through points F, E, D, C, B, and A. Repeat until a satisfactory curve is obtained.

(15) Stop the chart paper and switch the control panel selector to connect the top transducer (Position 2).

(16) Repeat Steps (9) through (14), this time turning the magnet on Fixture III and recording top CCW gear.

(17) Turn the master gear one index position clockwise (looking from top) so that the top master gear contacts the top CW gear on Fixture III, and the bottom master gear simultaneously mates with the bottom CW gear of Fixture VII.

(18) Repeat Step (15).

(19) Repeat Steps (9) through (14), this time turning Fixture III and recording top CW gear.

(20) Set the selector switch to Position 3 and repeat Steps (9) through (14), this time rotating Fixture VII and recording, the bottom CW gear.

(21) Continue indexing the master gear and recording the mated test gears until 180 deg total clockwise rotation is reached.

(22) Return the master gears to the neutral position.

(23) Record data on the remaining half of the gears with the master gear advanced in a CCW direction from the neutral position until 180 deg is again reached and return master gears to the neutral position.

Tabulation

With recorder settings as specified in the previous section, the output on the chart paper will be 100 mV/cm and tabulation must be done according to instructions in the section on measurement.

$$\text{overall measurement in millivolts} = \frac{AB + EF}{2} + CD$$

As the gears wear, both dimensions of the above equation will progressively increase and may be recorded separately and as a sum.

A battery is too short-lived to use it for the 6 V supply to the LVDT. Instead, a constant voltage power supply of high accuracy is recommended.

APPENDIX C
VENDORS

The vendors from whom important components or materials were purchased are listed below.

Gears (Involute)	Aero-Gear Machine and Tool Corp. 74 Industrial Avenue Little Ferry, N. J.
(Cycloidal)	Gear Specialities Co. 2635 West Medill Avenue Chicago, Ill.
Phosphor-Bronze (Filled with MoS_2)	S-K-C Research Associates 1 Thomas Road South Hawthorne, N. J.
Bearings	New Hampshire Ball Bearing Co. Peterborough, N. J.
Motors (Type 2308, 24 V dc)	Indiana General Corp. Electro Mechanical Division Ogelsby, Ill.

APPENDIX D

WEAR

Wear is a phenomenon which is characterized by the deterioration of solid surfaces when exposed to mechanical and/or chemical environments. Degradation of mechanical performance is usually the controlling criterion. Several theories, or rather hypotheses, concerning the phenomenon of wear exist. Experimental investigations have necessarily been conducted with relatively simple systems, usually pure metals, not without their own complexity. Even when the investigation is carried on in an ultrahigh-vacuum system the exposed surface differs considerably from the subsurface material.

In any other environment the surface is exposed to the ambient atmosphere. If the atmosphere is laboratory ambient then the surface is exposed to oxidation in the pressure of water vapor. A seemingly inert nitrogen atmosphere can permit an iron surface to become nitrided under some circumstances.

A survey of the existing literature indicates that the following mechanisms of wear are generally accepted for purposes of analysis: adhesive wear, abrasive wear, and surface fatigue. These classifications have been used for convenience of study and for unique representation of the wear process in controlled experiments where artificial boundaries are imposed for simplification or to study the effect of a particular parameter.

Adhesive wear can be analyzed in terms of the bulk properties of the materials, i.e., yield strength, modulus of elasticity, apparent area of contact and the normal load in the manner of Bowden and Tabor (ref. 5). It can also be investigated in terms of more parameters. Rabinowitz presented an analysis based on the surface energies of the materials (ref. 6). Both analyses permit one to gain very meaningful insight into the friction between material surfaces with relative motion. However these theories cannot be used to predict wear.

Rubbing can change the bulk physical properties. The temperatures of the contacting asperities, not the bulk temperature, must be used to determine the local effects on the bulk properties. Both the work hardening of the material and the change in contact stress due to wear induced changes in the surface profile must be known or accounted for in the analysis.

Of the materials selected for this study, 440C stainless steel, nitrided nitralloy, and carburized C1020 are the only ones used in which a matrix of one basic metal (iron) contacts. With the other materials a metal film coating of one type or another intervenes between the contacting matrices.

One copper based material (beryllium copper) was included. As would be predicted by either of the mentioned theories, copper does not have much wear resistance. The material combination which shows the greatest resistance to wear -- both in vacuum and air testing -- is the 440C through-hardened stainless steel and nitrided nitralloy. Both materials have a high yield stress and are quite hard.

The surface energy analysis cannot be applied because these energies of the two materials are not known. The fact that so little fundamental data is available for complex systems furthers the main difficulties in building an analytical model. We can, however, look at the range of wear expected using bulk properties.

Let us now consider the formulation of a quantitative adhesive wear model for determining the amount of material removed during the sliding action. This model was initially developed by Burwell and Strang (ref. 7) and the discussion here is from that work. It should be pointed out that little experimental verification exists for the application of this empirical model directly to mechanical components; however, from tests on materials the model appears valid under certain conditions (discussed in ref. 7) and the extension to our purposes is logical. In addition, this work represents the best quantitative wear model available in the literature reviewed to date.

In applying the shearing concept of Bowden and Tabor, it is recognized that the true area of contact between the solid surfaces is in reality only a fraction of the apparent contact area. Its value can be expressed as:

$$A = W/p_m$$

where

A = true contact area,

W = normal load,

p_m = the yield pressure producing plastic deformation of the softer material.

The volume of material removed, can now be expressed as

$$V = BAL$$

where

V = volume of material removed,

L = sliding distance

B = a factor expressing the probability of removing a single atom of material.

Upon eliminating the area of contact, this can be rewritten as

$$V = \frac{BWL}{p_m} \tag{1}$$

Expressing the amount of wear in terms of depth of material removed,

$$h = \frac{BPL}{p_m} \tag{2}$$

where

h = average depth of material removed,

P = average normal stress over the nominal contact area.

In obtaining equation (2) the value of P is obtained from

$$P = \frac{W}{A_o}$$

where

A_o = the apparent or nominal area of contact between the wearing surfaces.

Experimental data show that the value of B in equations (1) and (2) ranges between 1×10^{-7} and 5×10^{-7}. In addition, the value of p_m must be modified by a factor which depends upon the shapes of the asperities on the surface of the material. Bowden and Tabor suggest that the value of this factor will have a range between 1 and 3.

For a torque load of 20 oz-in. the normal stress is ~40 000 psi and if we assume the following approximate values,

$$N = 1800 \text{ rpm}$$

$$n = 55 \text{ teeth}$$

$$B = 5 \times 10^{-7}$$

$$p_m = (3)\ 180\ 000 \text{ psi}$$

$$L = \pi d \frac{N}{n} \times \Delta t = \frac{\pi\ 1.125}{56} (1.800/60) \times 100 \times 60 \times 60$$

$$\Delta t = 720 \text{ hr}$$

then the depth of material removal is

$$h = 18 \times 10^{-2} \text{ in.}$$
$$\approx 200 \times 10^{-3} \text{ in.}$$

and the actual wear is 9.8×10^{-4} in. or $\sim 10^{-3}$ in. There is a ratio of over two orders of magnitude between the actual wear and that predicted by the formula. Of course we must remember that the gears exhibit a combination of rolling and sliding.

In the air test the 440C stainless steel surface is protected by an oxide layer; however, the wear in the air test is greater than that of the vacuum test. Except for the general comparison of a harder, stronger material resisting wear relatively well (440C stainless steel vs nitrided nitralloy) and a softer material (beryllium copper) having poor wear resistance, there is little correlation on the basis of adhesive wear. Note that iron (nitrided nitralloy vs both 440C stainless steel and beryllium copper) has the expected resistance to wear (Figure 25).

In the abrasive type of wear, solid material is removed from a surface by being ploughed or gouged out by a much harder surface. This sort of wear is encouraged when the hard surface is a third body, generally a small particle of grit or abrasive. The shape of these wear particles is important and it has been shown (ref. 8) that angular particles of a soft material produce more wear than round hard particles. It has also been pointed out (ref. 9) that a good measure of the resistance to abrasive wear is the amount of elastic deformation that the surface can sustain. The larger the elastic limit of strain, the better the surface should be able to resist damage by an abrasive or harder surface. The following relationship can be stated,

$$E_{lim} = \frac{E_\sigma}{E}$$

where

E_{lim} = elastic limit of strain

E_σ = elastic limit of stress

E = material modulus of elasticity.

For a wide range of materials $E_\sigma \simeq H$, the indentation hardness. This can be restated as

$$E_{lim} \simeq \frac{H}{E}$$

and since the elastic strain energy per unit volume is

$$w = \frac{E_\sigma^2}{2E}$$

and

$$E_{\sigma} \simeq H$$

$$w \simeq \frac{H2}{2E}$$

hence

$$E_{1im} \simeq \frac{2}{H} \quad w$$

is derived. From these relations it can be concluded that, qualitatively, the abrasive wear resistance should vary directly with hardness and inversely with the elastic modulus.

This discussion of adhesive and abrasive wear suggests that the wear process is strongly influenced by the hardness of the material. This should be given closer examination, since on the basis of the preceding criteria, an increase in hardness from Rc34 to Rc60 would only increase,by approximately two,specimen wear resistance. Such a change in wear rate will in most cases be unnoticeable. However, we must recall that to increase the hardness of a material we have in all probability changed the chemical structure of the material. Therefore, as with carbon steels, the structure can change from iron to an iron carbide structure which will be more inert and have a lower surface energy. Such changes are likely to be of greater benefit than increases in hardness.

When rubbing or sliding surfaces are immersed in an environment which is corrosive or oxidizing, both adhesive and abrasive mechanisms tend to produce continuous removal of the products of corrosion or the oxides and thus lead to a general acceleration of wear. When the metal is subjected to cyclic stresses, materials of a high energy state are created continuously along the gliding planes of the crystals. The surface film on the material may also break down. Under the action of the stress and strain, however, some self-repair of this film is apt to occur.

Except for the beryllium copper gears, the wear in this study is generally indicative of abrasion. According to the criteria discussed in the preceding, the gears should have been much more resistive to abrasion. The qualitative explanation for the lack of wear resistance must lie in the combination of hard case and relatively soft matrix. A microscopic view reveals the surface to be comprised of peaks and valleys.

Abrasive degradation of the surface is the process that usually governs wear. The higher asperities are reduced, with concurrent high friction, and this is followed by a lower friction and lower wear rate operation. This is the process which seems to have occurred with most of the gear sets. The process is very dramatically seen in the case of the MoS_2 lubricated gears, in which a minor amount of debris is deposited during the first few hours, a long quiescent period of little or no debris formation follows, and the last period is characterized by extreme debris production. The following hypothesis explores this sequence.

During the wear period the crests of the asperities are fractured. In most cases this fracturing does not have a significant effect on the remainder of the hard coat, however, the base of the hard coat is weakened at some points. The larger the asperities, the more the hard coat is weakened.

During the quiescent period the weak points are being repeatedly stressed. At some time the weakened section fractures, leaving an unsupported section of the hard coat which then begins chipping off. Rapid wear follows.

The end of the quiescent period is signaled by rapid debris formation. With the Martin hard coated 7075T6 aluminum alloy gear the wear is very high, while in the others the wear is sufficiently low to leave part of the lubricating film on the gear tooth.

The rapid wear in the case of the unlubricated gears is similar. The soft matrix rapidly wears after the hard coat is chipped, finally leaving the coat as an incompletely supported shell. The shell cracks, being incapable of supporting a tension load, and the wear process continues. It probably would be worthwhile to run the gears under a light load for a wear-in period, then remove and relubricate them.

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